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January, 1935

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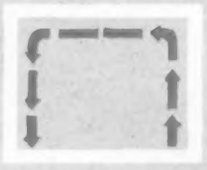

Directory of Industrial and Commercial Air Conditioning Equipment

American Society of Heating and Ventilating Engineers Journal Section

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VITAL AIR...

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Heating · Piping and Air Conditioning

January, 1935 *Vol. 7, No. 1*

Total Installed Air Conditioning Tonnage Increases 21% in 1934

Figures for 13 cities show total installed tons of refrigeration for air conditioning increased during 1934 from 65,000 to 78,500 tons

NINETEEN THIRTY-FOUR was "an air conditioning year." Installations of air conditioning systems last year (first six to eleven months—see table) in 13 cities[1] increased total installed tonnage by 21 per cent—in other words, 17½ per cent of the tonnage for air conditioning now installed in these cities went in to service just last year. Tons of refrigeration installed in all the years prior to 1934 amounted to 64,841; 13,673 tons were installed in 1934, making the present total 78,514. Using a figure of approximately $500 per ton as the cost of a complete, installed air conditioning system, the 1934 installations made in 13 cities alone would amount to nearly $7,000,000.

Residential air conditioning with mechanical refrigeration as yet amounts to a very small part of the total. For 6 cities[2] for which the data are available the tonnage for residential jobs totals 212, as compared with 40,171 tons for industrial and commercial installations.

The number of 1934 air conditioning installations in all classes of buildings substantiates the belief of many that 1935 is to be even more of an air conditioning year. It has been pointed out before that one air conditioning system leads to another, a fact well proved in the theater and railway fields. Air conditioning, at first a business advantage, rapidly becomes a competitive necessity as more and more stores, buildings, restaurants and other businesses acquire installations.

The data upon which the above conclusions are based (and which are summarized in the following paragraphs) have been obtained through the electrical util-

[1]Atlanta, Baltimore, Birmingham, Boston, Chicago, Cincinnati, Cleveland, Detroit, Kansas City, Manhattan, Nashville, New Orleans and Omaha.

[2]Atlanta, Birmingham, Chicago, Cincinnati, Detroit (mechanical compressor jobs only), and Nashville.

Summary of Air Conditioning Installations in 13 Cities

City	Prior to 1934			1934			Present Total		
	No. Jobs	Tons	Hp	No. Jobs	Tons	Hp	No. Jobs	Tons	Hp
Atlanta (8/30)	19	*2,560*	3,852	47	*300*	450	66	*2,860*	4,302
Baltimore (9/1)	...	*2,475*	3,692	60+	*1,165*	1,750	...	*3,640*	5,442
Birmingham (Nov.)	...	403	494	...	64	103	...	467	597
Boston (12/1)	...	*3,120*	4,682	48	*1,830*	2,723	...	*4,950*	7,405
Chicago (11/1)	379	20,267	35,210	228	4,970	5,845	607	25,237	41,055
Cincinnati (9/19)	75	*2,780*	4,161	73	*875*	1,310	148	*3,655*	5,471
Cleveland (10/1)	...	3,400[b]	3,825[b,c]	...	600	675	...	4,000[b]	4,500[b]
Detroit (Dec.)									
Compressor	121	*7,250*	10,889[a]	60	*476*	714[a]	181	*7,726*	11,603[a]
Steam Jet	1	14	...	1	167	...	2	181	....
Ice	9	61	...	...	...	...	9	61	....
Well Water	2	101	...	9	735	...	11	836	....
Kansas City (11/23)	96	1,668	2,493	149	1,059	1,029	245	2,727	3,522
Manhattan (9/1)	271	17,260	31,168	71	782	1,278	342	18,042	32,446
Nashville	8	*275*	410	6	*63*	95	14[c]	*338*	505
New Orleans (7/1)	38	2,347	4,092	13	207	366	51	2,554	4,458
Omaha (12/21)	16	*860*	1,287	38	*380*	567	54	*1,240*	1,854
Total	...	64,841	...	...	13,673	...	...	78,514	....

[a]Compressor hp only. [b]Estimated. [c]Does not include residential jobs totaling 20 hp.

NOTE: Tonnage figures shown in *italic* are estimated on basis of 1.5 hp = 1 ton. Dates after cities show part of 1934 included in compilation.

ities in each of the cities; this cooperation is gratefully acknowledged.

Baltimore—Over sixty installations were made in Baltimore between Jan. 1 and Sept. 1, 1934, the total horsepower put in service or contracted for being about 1,750 hp; this compares with a total of 3,692 hp for air conditioning at the end of 1933. Department stores (9 installations) lead with 753½ hp; 11 theater jobs total 595½ hp; and 6 miscellaneous installations (beauty shop, meat market, drug stores, and candy or confectionery stores) total 126½ hp. Six restaurants total 89½ hp; 15 installations classified under offices, clubs and institutions total 82 hp (mostly offices); while 5 specialty shops contribute 79 hp. Four residential installations total 3½ hp, three of them each being ½ hp and the fourth 2 hp. To the above figures should be added several residential and office installations not reported. One of the department stores included in the above with an electrical load of 126 hp uses district steam service in addition to this power requirement.

Boston—Forty-eight installations are reported for Boston during 1934 (to Dec. 1), and they represent a total horsepower of 2,723. Total hp prior to 1934 was 4,682. Twenty-three of the installations are central type, 23 are unit, and of the other 2, one is a theater using 7 tons of ice daily and the other represents New York, New Haven & Hartford R. R. car cooling, using 600 tons of ice per week. The unit type jobs represent a total of 105½ hp, and the central type 2,617½. One of these is a department store with 2,000 hp. The installations made in 1934 represent also restaurants, hotel bars and grill rooms, a hotel office, beauty shops, retail stores, offices, doctors' offices, banks, an auditorium, a theater, and residential.

Birmingham—Sixty-four tons of refrigeration are reported installed during 1934 (to Nov.), 53 for stores and commercial, 8.5 for offices, and 2.5 for residences. The store and commercial jobs represent a connected load of 76.7 hp; the offices 11 hp; and residential, 3 hp. Washed air systems for theater installations add 12 hp, making a total of 102.7 hp. Prior to 1934, there were 403.5 tons and 494 hp (theaters, 275 tons and 270 hp; stores and commercial, 105 tons and 183 hp; offices, 4.5 tons and 6 hp; and residences 19 tons and 35 hp) which added to the '34 jobs totals 467.5 tons and 596.7 hp for air conditioning in this city.

Chicago—During the first ten months of 1934, 228 installations were made, raising the total number of installations in Chicago proper from 379 to 607. The 1934 jobs represent 4,970 tons and 5,845 hp; as 20,267 tons and 35,210 hp were represented by the 379 installations prior to 1934, air conditioning in Chicago as of Oct. 31 amounted to 25,237 tons and 41,055 hp.

The largest 1934 tonnage was for theaters (1,385), 34 jobs being installed—20 with refrigeration and 14 being washed air systems. The next largest classification was general offices and buildings (1,037 tons), and third place goes to stores (957 tons). Twenty-three residential installations represented 39½ tons and 43½ hp. Seventeen industrial applications added 552 tons and 768 hp.

An accompanying table summarizes the number of installations, tons of refrigeration, and horsepower for the installations prior to 1934, those made in 1934, and the total as of Oct. 31, 1934, for various types of buildings. The large increase in tonnage made during the past year is indicative of considerable activity for 1935, as improved business conditions and the fact that "one air conditioning installation leads to another" should influence even greater increases this year.

Cincinnati—A total of 148 installations and 5,471.25 hp are reported (Sept. 19), 73 of which representing 1,309.75 hp were made in 1934. They are classified as follows:

	PRIOR TO 1934		1934		TOTAL	
	NO.	HP	NO.	HP	NO.	HP
Residences	14	27	25	35.25	39	62.25
Offices	18	54	25	127	43	181
Small Stores	5	45.5	10	221	15	266.5
Large Stores	2	405	2	640	4	1045
Restaurants	12	250.5	4	49	16	299.5
Theaters	9	1637.5	3	177.5	12	1815
Hospitals	4	26	1	19	5	45
Hotels (inc. restaurants)	3	1321	—	——	3	1321
Miscellaneous	8	395	3	41	11	436
Total	75	4161.5	73	1309.75	148	5471.25

Kansas City—The summary sheet for 1934 installations as of Nov. 23 shows a total of 149 jobs with a tonnage of 1,059¼ and 1,029 total hp. A classification of all installations in Kansas City as of Nov. 23, 1934, which includes the 1934 jobs shows a grand total of 245 installations, a tonnage of 2,727¼, and a total horsepower of 3,522⅔. Previous to 1934 there were, therefore, 96 installations with a total tonnage of 1,668 and a total hp of 2,493⅔.

Philadelphia—A total of 131 installations were made to Aug. 31, consisting of 82 commercial, 12 industrial, and 37 residential. The total connected horsepower for the 131 installations is 2,188.8, of which 1,712.5 is for commercial buildings, 420 for industrial, and 56.3 for residential. The commercial group includes 5 apartments, 3.5 hp; 1 bank, 140 hp; 24 offices, 122 hp; 1 funeral parlor, 12 hp; 2 hospitals, 17 hp; 1 church, 6 hp; 1 museum, 10

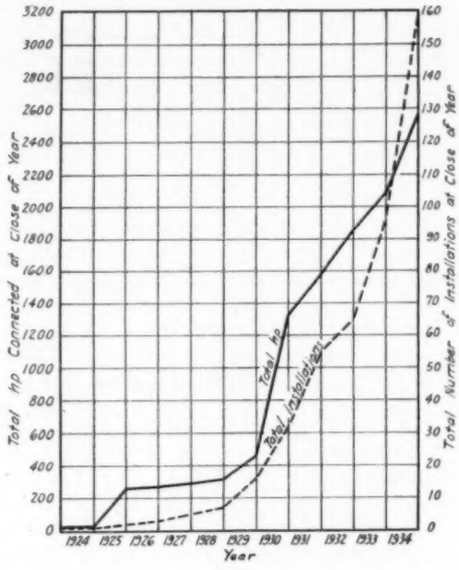
Summary of Air Conditioning Installations, Chicago

	PRIOR TO 1934			1934			TOTAL		
	No. Jobs	Tons	HP	No. Jobs	Tons	HP	No. Jobs	Tons	HP
Banks	5	741	1,206	3	172½[a]	46½	8	913½	1,252½
Brokers & Exchanges	6	490½	721	2	35	41	8	525½	762
Beauty Shops	1	4½	4½	4	37	63¾	5	41½	67¾
Churches	2	65	95	1	60	65	3	125	160
Civic Bldgs.	1	50	116½	0	0	0	1	50	116½
Clubs	4	202	359	0	0	0	4	202	359
Dance Halls	4	325	465	0	0	0	4	325	465
Dentists	2	1½	1½	2	2	2	4	3½	3½
Drug Companies	2	32	56	5	81	104¼	7	113	160¼
Hospitals	2	27	33	5	8	8½	7	35	41½
Hotels	16	1,605½	3,282	6	100	124	22	1,705½	3,406
Industrial	61	2,828½	5,767	17[b]	552	767¾	78	3,380½	6,534¾
Gen. Off. & Bldgs.	27	3,258½	4,758½	24	1,037[c]	926	51	4,295½	5,684½
Private Offices	60	100	131	49	114½	131½	109	214½	262½
Residences	21	26½	29	23	39½	43½	44	66	72½
Restaurants	63	1,361	2,113½	25[d]	346½	443½	88	1,707½	2,557
Stores	27	995½	1,436½	22[e]	956¾	1,137¾	49	1,952¼	2,574¼
Studios, Radio	2	260	355	1	10	12	3	270	367
Undertakers	5	26	60	5	32½[f]	42¼	10	58½	102¼
Theaters	68[h]	7,867	14,220	34[i]	1,385	1,886	102[g]	9,252	16,106
TOTAL	379	20,267	35,210	228	4,969¼	5,844¾	607	25,236¼	41,054¾

[a]Includes one 150-ton steam jet.
[b]Includes one 250-ton, 300 hp car cooling installation.
[c]Includes one 600-ton 300 hp. steam jet.
[d]One job on central hotel refrigerating plant and 6 with no compressor.
[e]Includes two on central refrigerating plant.
[f]Includes one with no compressor.
[g]68 with refrigeration, 34 washed air only.
[h]48 with refrigeration, 20 washed air only.
[i]20 with refrigeration, 14 washed air only.

hp; 14 restaurants, 208 hp; 12 specialty stores, 513 hp; 1 theater, 20 hp; 2 trains,[3] 280 hp; 20 schools,[4] 381 hp. The industrial group comprises 3 candy manufacturers, 290 hp; 2 chemists, 9 hp; 2 dairies, 26 hp; 1 printer, 1 hp; 1 storage, 30 hp; 1 cigar manufacturer, 12 hp; 1 textile, 22 hp; and 1 gold leaf manufacturer, 30 hp.

Pittsburgh—Fifty-six installations totaling 326 tons of refrigeration and 415 hp were made during 1934. The only installation over 50 hp is a 60 ton job for the first floor of the Duquesne Light Company's central office building.


Total hp Connected at Close of Year
Total Number of Installations at Close of Year
Total hp
Total Installations
Year
1924 1925 1926 1927 1928 1929 1930 1931 1932 1933 1934

Total connected air conditioning load and number of installations for the territory served by the Oklahoma Gas and Electric Co.

Northern California—In the territory served by the Pacific Gas and Electric Company, northern California, 79 residential installations for the first nine months of 1934 are on record; they total 39 hp in connected load. The 75 commercial and industrial installations made during this period have a total connected load of 195 hp.

Nashville—Six commercial installations during 1934 total 95 hp and 8 jobs installed prior to 1934 total 410½ hp (two are theaters of 150 and 100 hp respectively). Thus the record shows 14 commercial installations and 505½ hp as of November, 1934. Residence installations made during 1933-34 represent an additional 20 hp. The forecast for 1935 is 200 hp, probably averaging 10 hp per installation.

Cleveland—During the first nine months of 1934, the air conditioning installations sold amounted to 600 tons and 675 hp. It is estimated that with the jobs prior to 1934 there are 4,000 tons of refrigeration for air conditioning and a total connected hp of 4,500. Only 15 residential installations are reported for this city.

Atlanta (and suburbs)—Sixty-six installations totalling 4,302 hp are reported as of August 30, 47 of which totalling 450 hp were made in 1934. Four of the earlier jobs are theaters representing some 1,700 hp, one a store with a connected load of 631 hp, and one an industrial installation estimated as 1,500 hp. Twelve of the 66 installations (3 prior to 1934 and 9 during 1934) are residential and total 34 hp, the rest being commercial and industrial and including theaters, offices, stores and shops, restaurants, hotels, hospitals, etc.

Omaha—Thirty-eight installations totalling 567 hp are reported as of Dec. 21; as there were 16 jobs and 1,287 hp prior to 1934, total number of installations in Omaha is 54 and the total hp 1,854. Three of the theater installations prior to 1934 represented 358, 440, and 348 hp which accounts for the discrepancy between the number of jobs and total hp for 1934 and for prior to 1934.

New Orleans—There are 51 installations in New Orleans, 13 of which were made during the first 6 months of 1934—4 small stores, 1 large store, 3 small offices, and 5 restaurants. The 1934 jobs represent 366 hp and 207 tons. The 38 jobs prior to 1934 represent 5 theaters (1,412 hp and 670 tons), 4 small stores (82 hp and 35 tons), 5 large stores (1,885 hp and 1,140 tons), 1 large office building (516 hp and 350 tons), 3 large offices (31 hp and 20 tons), 6 small offices (10 hp and 7 tons), 2 broadcasting studios (24 hp and 11 tons), 1 brokers office (10 hp and 6 tons), 1 hotel (103 hp and 100 tons), and 11 industrial jobs (19 hp and 8 tons). Thus the total tons prior to 1934 was 2,347 and the total hp 4,092, which added to the 1934 figures result in 2,554 tons and 4,458 hp now installed.

Detroit—The tabulation below shows number of installations and hp prior to 1934 and installed during 1934; the hp figures given for the compressor installations are compressor hp and do not include fan and pump hp:

Air Conditioning Installations, Detroit

Compressor Installations

	Prior to 1934		1934	
	No.	HP	No.	HP
Offices	28	88½	12	99
Office Buildings	2	960	..	..
Restaurants	13	351	6	109
Night Clubs	2	15	2	25
Hotels	6	722	1	20
Residences	16	54	12	18
Department Stores	3	3,353	1	30
5c to $1 Stores	1	40	1	120
Furniture Stores	1	12	..	..
Candy Stores	5	26½	2	42½
Drug Stores	3	47½	2	9½
Jewelry Stores	3	17½	2	22½
Opticians	..	..	2	2
Clothing Stores	3	18	8	96
Miscellaneous Stores	1	20	..	..
Electric Company Sales Offices	2	34	1	7½
Radio Stations	1	1½	..	..
Theaters	16	3,670	1	45
Library	..	..	1	4
Banks	3	600	..	..
Stock Exchange	1	50	..	..
Broker's Room	1	75	..	..
Barber Shops	1	30	..	..
Beauty Parlors	1	6	1	5
Hospital	1	60	..	..
Laboratories	2	53	..	..
Drug Manufacturing	1	575	1	6
Candy Manufacturing	1	1	1	8¼
Printing	..	..	1	20
Undertaking Parlors	3	9	2	25
	121	10,888½	60	714¼

Steam Jet Installations

	Prior to 1934		1934	
	No.	Tons	No.	Tons
Restaurants	1	14	..	..
Hotels	..	..	1	167
	1	14	1	167

Ice Installations

	No.	Tons	No.	Tons
Retail Stores	4	30	..	..
Offices	3	17	..	..
Residences	1	6	..	..
Restaurants	1	8	..	..
	9	61		

Well Water Installations

	Prior to 1934		1934	
	No.	Tons	No.	Tons
Theaters	1	85	4	376
Night Clubs	1	16	..	..
Restaurants	..	..	1	66
Candy Stores	..	..	3	160
Miscellaneous Stores	..	.	1	133
	2	101	9	735

[3]Represents the half-hour demand on utility's system by trains loading in two terminals.
[4]Do not include mechanical refrigeration.

Michigan (except Detroit)—While the territory served by the Consumers Power Co. does not include the entire state, it is believed that most of the Michigan air conditioning installations outside Detroit are in its territory. Records are available on 27 installations made during 1934, and on 12 prior to Jan. 1, 1934. In the 1934 jobs are included the following:

Office, Battle Creek: Unit coolers with central comp., 30 tons ref., 1-15 hp and 1-10 hp comp. motors, 2-2 hp pump motors.

Cafeteria, Battle Creek: Unit type, approx. 48,000 gal of 55 F water used per month, 4-100 watt fan motors.

Theater, Flint: Central system, 55 gpm of water at 49 to 52 F used, 30,000 cfm fan, 1-10 hp pump motor.

Restaurant, Flint: Central system using 52 F water, 3,500 cfm fan, ¾ hp fan motor, 2 hp pump motor.

Office, Flint: 3 units—central, using 52 F water; fans are 14,000, 4,000, and 16,000 cfm; 3-5 hp fan motors, 1-20 hp pump motor.

Office, Grand Rapids: Central system, 6¾ tons, 2,750 cfm fan. 10 hp compressor motor, ¾ hp fan motor.

Office, Grand Rapids: Self-contained unit, 1 ton, 400 cfm fan, 1 hp compressor motor, 1/20 hp fan motor.

Office, Grand Rapids: Same as above.

Ready-to-wear fitting room, Grand Rapids: Unit, ¾ ton, 1 hp compressor motor, 1/20 hp fan motor.

Tea room, Grand Rapids: Central system, 75 gpm of 55 F well water, 35,000 cfm fan (used for heating much larger area in winter), 25 hp fan motor, 5 hp pump motor.

General offices, Grand Rapids: Central system, 30 gpm of 50 F water, 6,000 cfm fan, 2 hp fan motor.

Office, Grand Rapids: Central duct system, 50 gpm of 54 F water, 6,500 (plus) cfm fan, 5 hp fan motor (load 3.5).

Residence, Hastings: Central system using water as cooling medium.

Residence, Hastings: Same as above.

Residence, Hastings: Same as above.

Bedroom, Jackson: Unit, ⅓ ton, 300 cfm fan, ½ hp compressor motor, 1/100 hp pump motor.

Restaurant, Jackson: Unit system, 24,000 Btu per hr, 5 hp compressor motor.

Bedroom, Jackson: Unit, ¾ ton, 300-400 cfm fan, 1 hp compressor motor, 1/30 hp fan motor.

Office, Kalamazoo: Indirect unit using 25 gpm of 55 F water, 2,000 cfm fan, ½ hp fan motor.

Living and bed room, Kalamazoo: Self-contained unit, ½ hp compressor motor.

Offices, Kalamazoo: Central type using 58 F water, 14,770 cfm fan, 5 hp fan motor, 5 hp pump motor.

Office, Muskegon: Self-contained unit, 1 ton, 500 cfm fan, 1 hp compressor motor, 1/10 hp fan motor.

Bakery, Muskegon: Unit, 3.4 gpm of 55 F water, 3-speed fan—1,500 to 3,200 cfm, ¾ hp fan motor.

Hotel cafeteria, Muskegon: Central, 6 tons per hr, 16 F entering brine —40 F leaving, 2,650 four-speed fan, 1/6 hp fan motor.

Living room, dining room, and den, Pontiac: Unit with compressor in basement, 400 cfm fan, 1½ hp compressor motor, 1/20 hp fan motor.

Office and drafting room, Pontiac: Duct distribution, 42 gpm of 56 F water, 6,000 cfm fan, 2 hp fan motor.

Office, Saginaw: Unit, ½ ton compressor, 400 cfm fan, 1 hp compressor motor, 1/30 hp fan motor.

New York City (Manhattan only)—Seventy-one installations were made in 1934 to Sept. 1, totalling 782 tons and 1,278 hp, and 3,770,240 cu ft of air conditioned space. These figures compare with 271 installations prior to 1934, 17,260 tons, 31,168 hp, and 102,197,800 cu ft of conditioned space. The 1934 jobs include 2 banks (one a steam jet job), 1 beauty parlor, 2 hotels, 3 industrial, 25 offices, 5 residences, 17 restaurants, 7 stores, 1 radio studio, 2 theaters (one ice) and 6 miscellaneous. The restaurant installations account for 343 tons and 462 hp, and the offices for 291 tons and 560 hp, while at the other extreme are residences, the five installations accounting for 7 tons and 8 hp. Prior to 1934 there were 9 bank installations, 3 beauty parlors, 3 clubs, 3 doctors and dentists, 1 drug store, 2 hospitals, 18 hotels, 7 industrial, 80 offices, 15 residences, 53 restaurants, 21 stores, 3 radio studios, 43 theaters (of which 6 are water and 2 ice), and 10 miscellaneous.

Oklahoma and Arkansas—For the territory served by the Oklahoma Gas and Electric Company, which includes some 230 cities and towns in Oklahoma and Arkansas, 48 installations with mechanical refrigeration and 15 without mechanical refrigeration are reported for 1934 (to Nov. 15). The horsepower for the mechanical refrigeration jobs totals 350 and for the 15 evaporative cooling jobs, it is 133.5 hp. Thus there were a total of 63 installations in 1934 representing 483.5 hp. Previous to 1934 there were 28 mechanical refrigeration installations with 1,242.56 hp, 68 evaporative cooling installations with 833.75 hp—a total of 96 installations with 2,076.31 hp. A number of air conditioners which were in operation during the season are not included in the 1934 figures, either because they are to be replaced next year by larger equipment, the user moved to a new location, or decided not to buy. Several conditioners were sold but not installed because of a city water shortage which made it impossible to secure cooling water.

The trend in the number of installations and total horsepower in this territory as shown in an accompanying graph—and particularly the increases during 1934—lead to the conclusion that 1935 will offer an outstanding opportunity for the installation of air conditioning equipment.

Guests Like Comfort of Cooled Rooms, Detroit Statler Finds

"WE LIKE it and will tell our friends; it should bring much business to you"—this statement summarizes the reaction of guests of the Hotel Statler, Detroit, to the comfort cooling systems installed last summer for 29 guest rooms and for public spaces. The 151 favorable comments made to floor clerks, assistant managers and others coming in contact with guests included:

Perfectly pleased	93
Rooms most comfortable of any ever experienced	20
Enjoyed good night's rest	14
Relieved hay fever	4
Relieved asthma	1
Stopped at Statler because of air conditioning	4
Was free from dirt	2
Liked comfort of dining and ball-rooms	13

The comparatively few unfavorable comments, most of them made soon after the installation and before the conditions complained of were corrected, included "system not working," "too noisy," "not as much air flow as expected," "caught cold," "too stuffy," "too much difference between corridor and room temperature," etc.

During June, July, and August of last summer the room count increased 17 per cent and the number served in the dining rooms increased 24½ per cent over the previous three year average. While it would be difficult to say what part of the increase can be attributed to the air conditioning, the comparable figures for the company's Buffalo hotel are a decrease of 14 per cent in room count and an increase of 18.8 per cent in number of guests served in the dining room, and for the Hotel Statler in Cleveland, a decrease of 25.4 per cent in room count and 5.1 per cent in guests served.

Cost of operation (coal, current and water) of the air conditioning system was $844.50 for ten days in June, $1,547.90 for July, and $1,594.55 for August; total, $3,986.95. The refrigeration equipment is the steam jet type.

DESIGN OF MODERN INDUSTRIAL PIPING SYSTEMS —— the Flow of Fluids

By **F. L. Snyder***

DURING the past few years we have found that certain fluid flow formulas based on the volume of the fluid without considering its viscosity at operating temperature are inaccurate. The continued use of these formulas has not only resulted in the uneconomical use of pipe materials, but large power waste as well. In many instances initial installations that had been designed for future capacities were found to be insufficient when operated at the design capacity, while other installations could care for ultimate capacities greater than provided for by the original design. Investigation determined that in many cases these inaccuracies existed where the formula used was not corrected for the viscosity of the fluid. Other cases proved that the designer, although determining the proper size of the pipe, did not consider the pipe bends in the system as increasing the pressure loss through the line, assuming that a long sweep bend of about five pipe diameters or more did not increase the pressure loss, but that a welding fitting having a radius of 1.5 pipe diameters created a pressure loss equal to that of a standard screw or flanged type elbow. Experiments have disproved this theory.

These conditions prompted the author to investigate fluid flow formulas, and pressure losses through elbows and pipe bends. Therefore, the following data, substantiated by laboratory experiments which have been checked with actual operating conditions, are compiled. These data are primarily to assist the designer to construct more efficient pipe systems, yet reduce and simplify the effort required for such design in that all information pertinent to design of pipe, valves, fittings, etc., is combined. These data have been in constant daily use for the past four years and have not only given accurate results, but substantial savings have been gained through the economical use of pipe materials and the reduction of power waste. While laboratory experiments were made primarily to determine the authenticity of certain fluid flow formulas, the experiments made under actual operating conditions substantiate the fact that the fundamental Fanning formula is more nearly correct than others now used. This formula may be written in different forms, depending on the units employed. The following are typical:

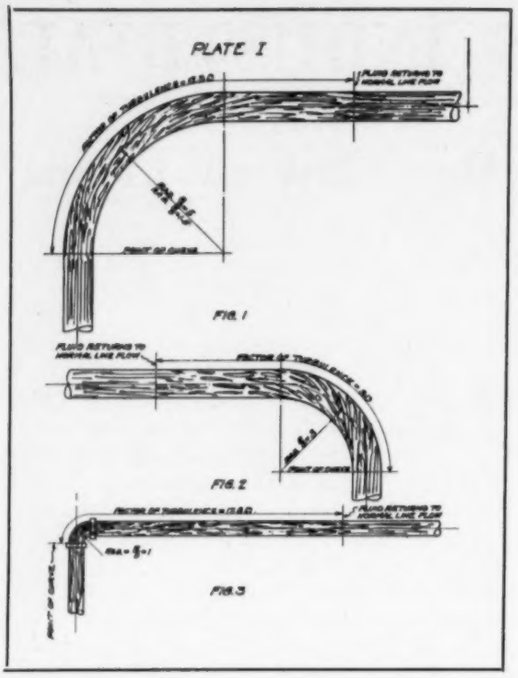
$$\Delta p = 0.322 LfSu^2/D$$
$$\Delta p = 0.00517 Lfwu^2/D$$
$$\Delta p = 13.4 Lfm^2/wD^5$$
$$\Delta p = 13.4 LfwQ^2/D^5$$

The Fanning formula is based on the density at working temperature, which affects the viscosity of the fluid. While the viscosity of water is little affected by a rise or fall in temperature, the viscosity of some fluids, such as oils, often varies through a wide range with a comparatively small rise or fall in the temperature. Therefore, the viscosity of the fluid at the extreme working temperature must be known. In many instances viscosities are referred to as Saybolt Universal, Engler, or Redwood time seconds. It is therefore necessary to convert these measurements into the common unit kinematic viscosity, which in turn is expressed in centipoise units Z, which varies as the specific gravity of the fluid. To simplify this conversion "Viscosity Conversion Chart No. 7" was constructed with formulas of W. H. Herschel (National Bureau of Standards technological papers No. 100 and No. 112). Assuming the fluid has a Saybolt Universal viscosity of 1,000 time seconds and that the specific gravity is 1; refer to "1,000 time seconds" on the Saybolt Universal viscosity scale and read directly opposite the kinematic viscosity "2.2." To convert kinematic viscosity to centipoise units multiply by 100 times the specific gravity or $2.2 \times 100 \times 1 = 220$ centipoises. Viscosity is also affected by extremely high pressures, but is little affected by pressure ordinarily used in commercial practice.

The value of the friction factor f in the Fanning formula is dependent on the Reynolds number and expressed as $f = m / ZD$. This equation when plotted against the Reynolds number forming a friction factor curve shows three definite breaks, the first occurs in the viscous or streamline flow region having a value of f up to 0.14; the second occurs in the turbulent flow region where f is greater than 0.14 and less than 27 and the third occurs in the turbulent flow region where f is 27 or greater.

Because of inaccuracies in certain fluid flow formulas which have resulted in uneconomical design of industrial piping systems, the author investigated.....substantiated his findings by laboratory experiments ..... checked them under actual operating conditions.....presents his results here. This information has been in constant daily use, has led to accurate results, made possible substantial savings in economical use of pipe materials and in reduction of power waste.

*Engineering Department, E. I. du Pont de Nemours & Co., Wilmington, Del.



Throughout this article the following symbols are used:

Δp = Pressure drop in lb per sq in.
Δp_e = Pressure drop in lb per sq in. due to increase in diameter of the line carrying the fluid.
Δp_e = Pressure drop in lb per sq in. due to decrease in diameter of the line carrying the fluid.
Δp_r = Pressure drop in pipe coils, lb per sq in. per ft of pipe.
f = Friction factor dependent on Reynolds number m/ZD.
f_1 = Roughness multiplication factor.
f_2 = Constant determined with Chart No. 6.
L = Length of pipe, ft.
D = Inside diameter of pipe, inches.
R_c = Radius of pipe coils, inches.
a = Inside area of pipe transverse section, sq ft.
S = Specific gravity referred to water = 1.
u = Velocity in feet per second at density w.
w = Density in lb per cu ft at arithmetic mean pressure equals 62.4 times S.
M = Thousands of lb per hr per sq ft of pipe transverse section area.
m = Thousands of lb per hr.
Q = Thousands of cu ft per hr at density w.
k = Kinematic viscosity.
Z = Viscosity in centipoise units.
g = Acceleration of gravity = 32.16.
SU = Saybolt Universal time, seconds.
E = Engler time, seconds.
Rw = Redwood time, seconds.

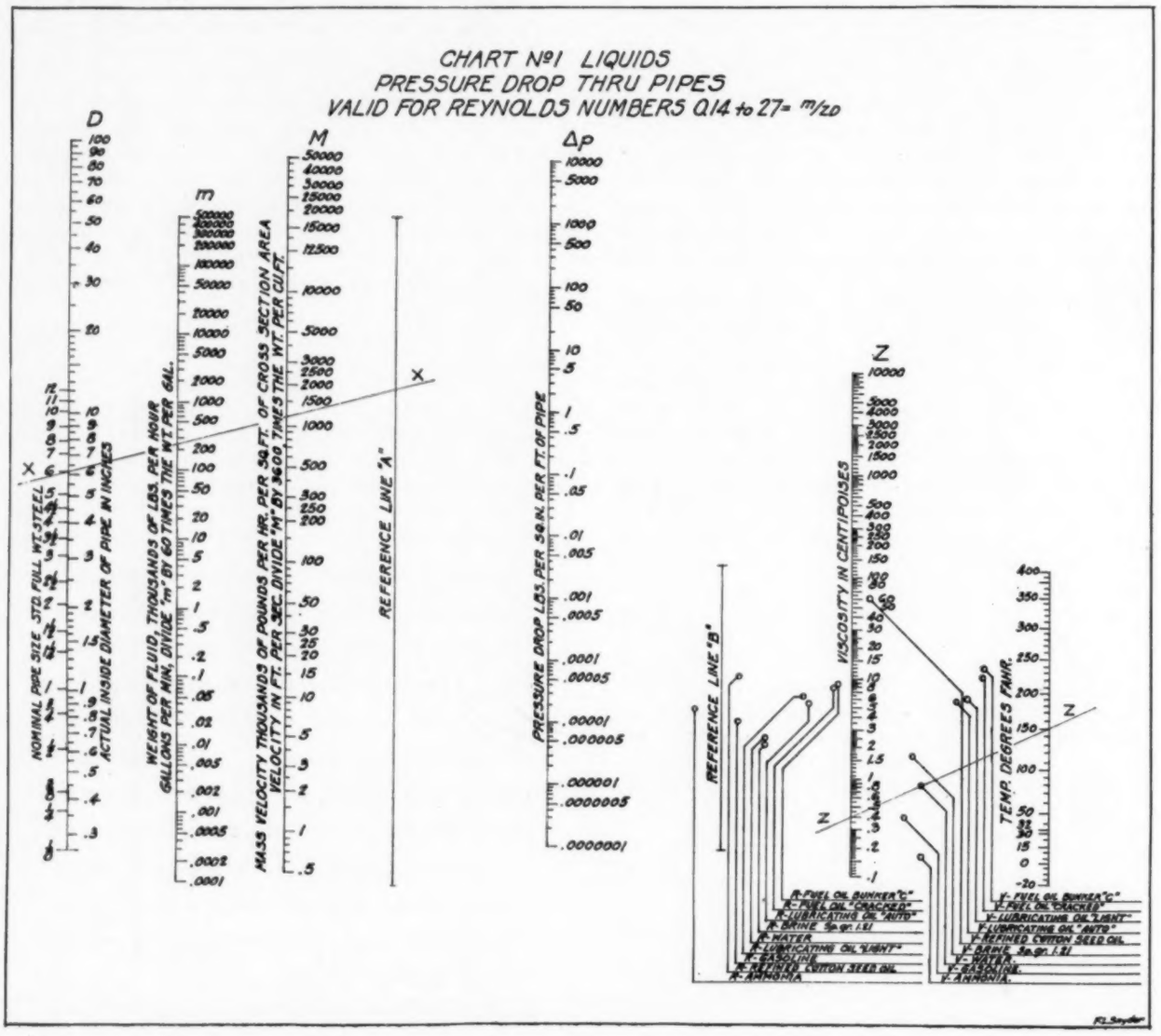


CHART Nº1 LIQUIDS
PRESSURE DROP THRU PIPES
VALID FOR REYNOLDS NUMBERS 014 to 27= m/zD
D
NOMINAL PIPE SIZE STD FULL WT STEEL
ACTUAL INSIDE DIAMETER OF PIPE IN INCHES
m
WEIGHT OF FLUID, THOUSANDS OF LBS. PER HOUR
GALLONS PER MIN. DIVIDE "m" BY 60 TIMES THE WT PER GAL.
M
MASS VELOCITY THOUSANDS OF POUNDS PER HR. PER SQ FT. OF CROSS SECTION AREA
VELOCITY IN FT. PER SEC. DIVIDE "M" BY 3600 TIMES THE WT. PER CU FT.
REFERENCE LINE "A"
Δp
PRESSURE DROP LBS. PER SQ IN. PER FT. OF PIPE
Z
VISCOSITY IN CENTIPOISES
REFERENCE LINE "B"
TEMP. DEGREES FAHR.
A- FUEL OIL BUNKER "C"
B- FUEL OIL "CRACKED"
C- LUBRICATING OIL "AUTO"
D- BRINE Sp. gr. 1.21
E- WATER
F- LUBRICATING OIL "LIGHT"
G- GASOLINE
H- REFINED COTTON SEED OIL
I- AMMONIA
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Where stream line flow is encountered, Poiseuille's formula is applicable:

$$\Delta p = 0.000668 L f Z u / D^2$$
$$\Delta p = 0.034 L f Z m / w D^4$$

The majority of pressure drop problems, however, occur in the turbulent region of the friction factor curve, and in order to simplify the computation necessary for pipe size design, nomographic charts Nos. 1 to 4, based on the Fanning formula, have been constructed. Charts Nos. 1 and 2 are for liquids, Charts Nos. 3 and 4 for gases. The use of the charts is explained below.

Since the Fanning formula is based on new steel pipe, it must be supported by the roughness multiplication factor f_1. After a pipe line has been in service for any length of time the inside surface of the wall may undergo a mechanical change caused by erosion or corrosion, which increases the friction, incidentally increasing the pressure drop. If the pipe is made of corrosive or erosive resistant materials this factor need not be considered. It is suggested that consideration be given thin wall pipe made of corrosive resisting alloy and installed with welded joints. In many instances the pressure at which the fluid is flowing is not in proportion to the pipe wall thickness due to the present-day pipe standards, which include a thread depth safety factor. This may result in the justification of a thin wall or lined pipe.

Experiments on pipe that have been in service a number of years resulted in the use of the roughness multiplication factor f_1 by Hazen-Williams. Therefore, the pressure drop as determined with the nomographic charts or formulas should be multiplied by any of the following factors:

CONDITION OF THE PIPE

	f_1
Smooth, brass, copper or lead pipe	0.90
New steel, cast iron or corrosion resisting pipe	1.00
Smooth wooden or good concrete lined pipe	1.20
Old cast iron or new riveted steel pipe	1.40
Vitrified or old steel pipe	1.60
Old riveted steel pipe	2.00
Badly turberculated pipe	2.50
General use (a good safety factor for ordinary conditions)	1.20

It has been determined that the ideal radius of a bend so far as pressure drop is concerned is between 2.5 and 3 pipe diameters and that the pressure drop increases as the radius of the bend is increased or decreased from this value. As the radius increases above three pipe diameters, the pressure drop increases, due (apparently) to the time factor of turbulence in making the direction

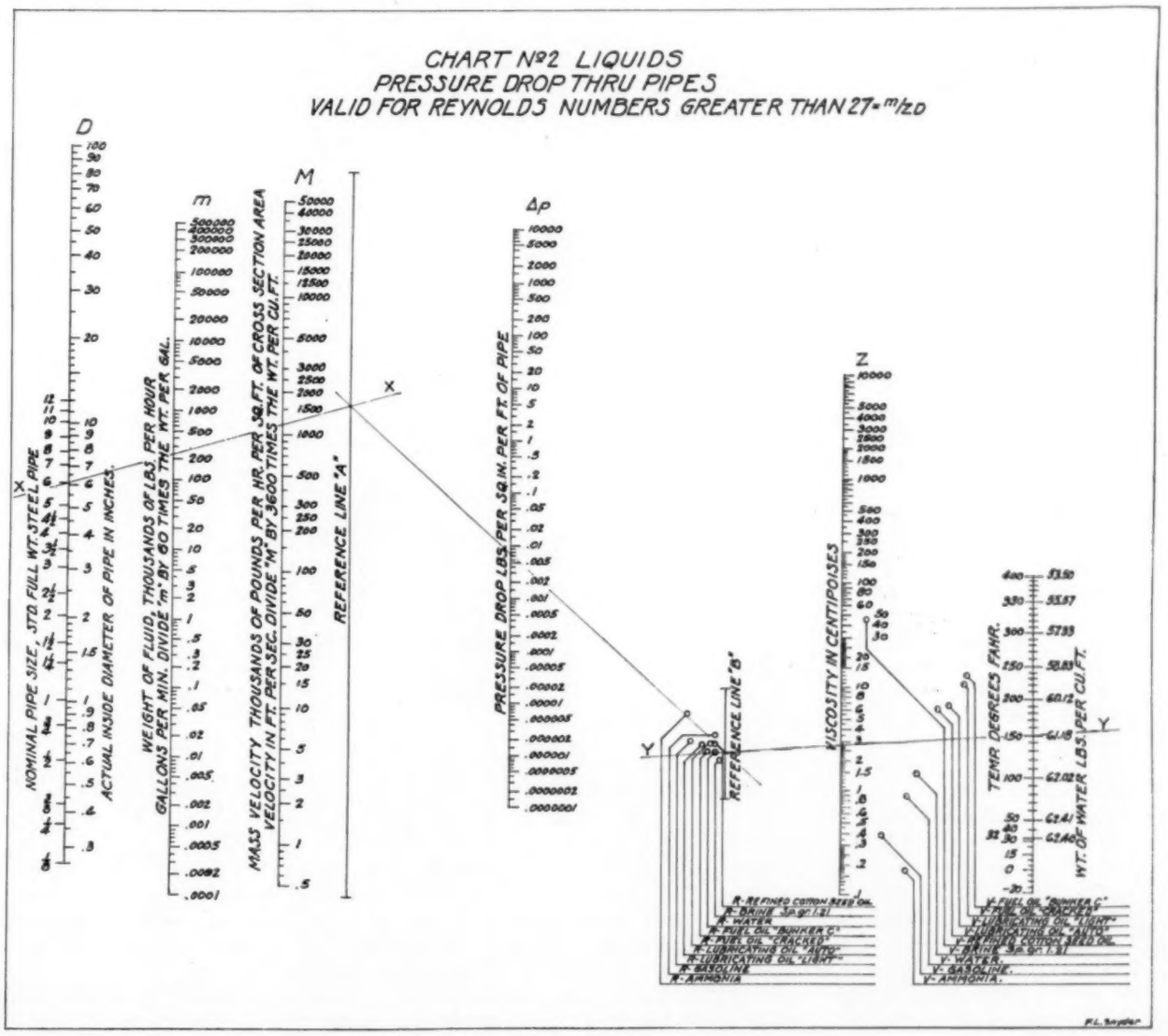


CHART Nº2 LIQUIDS
PRESSURE DROP THRU PIPES
VALID FOR REYNOLDS NUMBERS GREATER THAN 27·m/zD
D
NOMINAL PIPE SIZE, STD. FULL WT. STEEL PIPE
ACTUAL INSIDE DIAMETER OF PIPE IN INCHES.
m
WEIGHT OF FLUID, THOUSANDS OF LBS. PER HOUR
GALLONS PER MIN. DIVIDE "m" BY 60 TIMES THE WT. PER GAL.
M
MASS VELOCITY THOUSANDS OF POUNDS PER HR. PER SQ. FT. OF CROSS SECTION AREA
VELOCITY IN FT. PER SEC. DIVIDE "M" BY 3600 TIMES THE WT. PER CU. FT.
REFERENCE LINE "A"
Δp
PRESSURE DROP LBS. PER SQ. IN. PER FT. OF PIPE
REFERENCE LINE "B"
Z
VISCOSITY IN CENTIPOISES
TEMP. DEGREES FAHR.
WT. OF WATER LBS. PER CU. FT.
X
Y
R-REFINED COTTON SEED OIL
R-BRINE Sp. Gr. 1.21
R-WATER
R-FUEL OIL "BUNKER C"
R-FUEL OIL "CRACKED"
R-LUBRICATING OIL "AUTO"
R-LUBRICATING OIL "LIGHT"
R-GASOLINE
R-AMMONIA
V-FUEL OIL "BUNKER C"
V-FUEL OIL "CRACKED"
V-LUBRICATING OIL "LIGHT"
V-LUBRICATING OIL "AUTO"
V-REFINED COTTON SEED OIL
V-BRINE Sp. Gr. 1.21
V-WATER
V-GASOLINE
V-AMMONIA

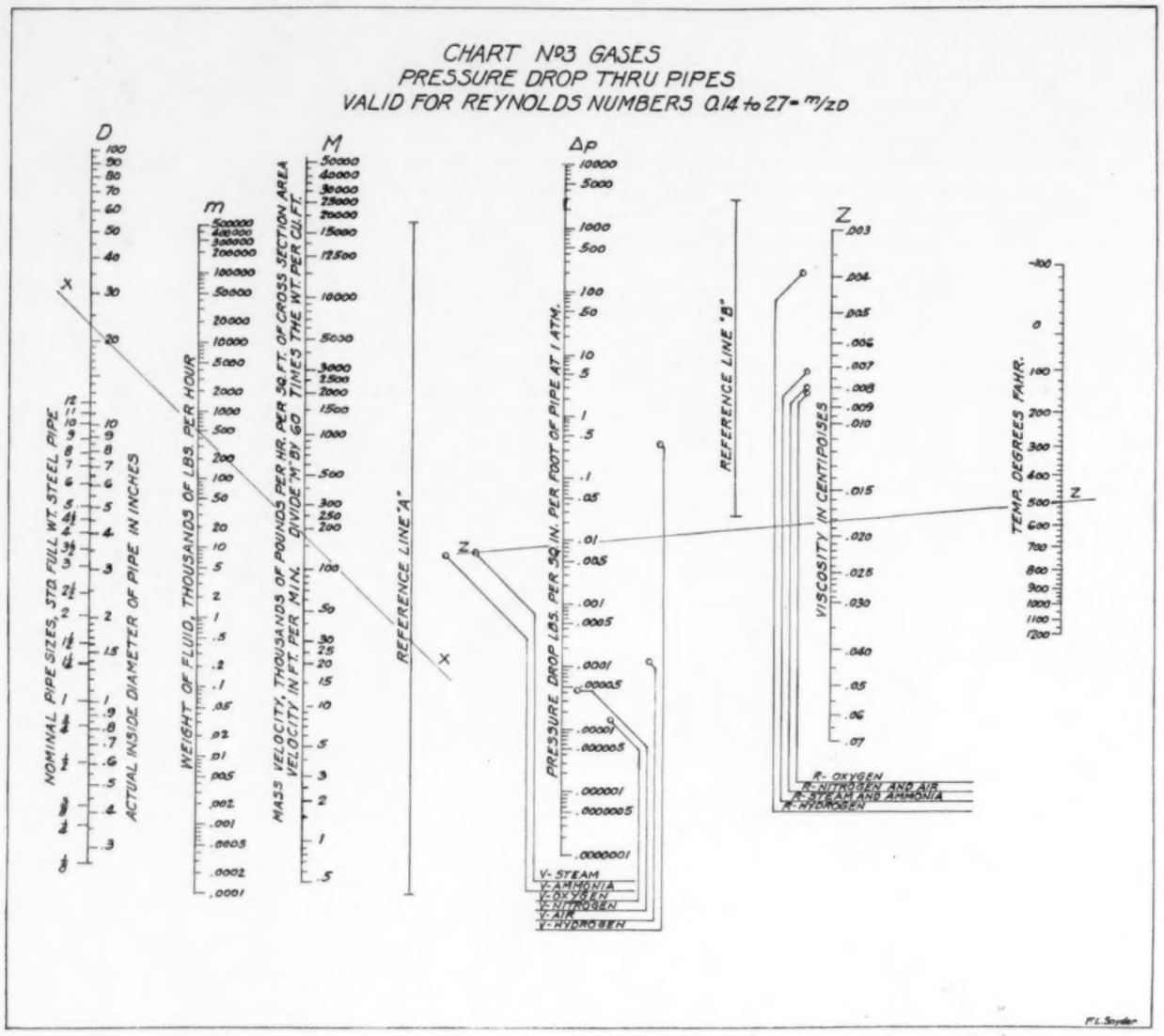


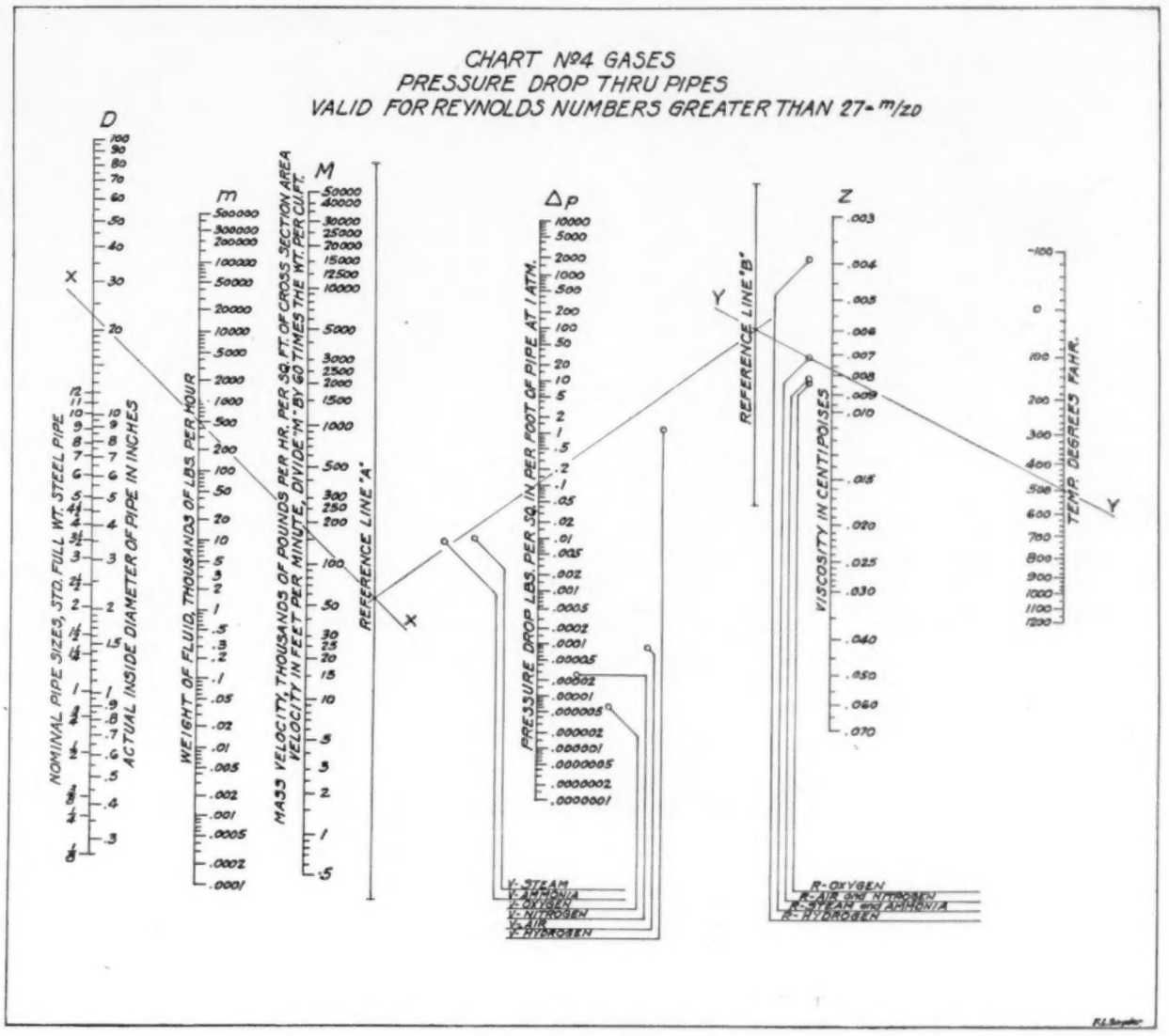
CHART Nº3 GASES
PRESSURE DROP THRU PIPES
VALID FOR REYNOLDS NUMBERS 0.14 to 27 - m/zD
D
NOMINAL PIPE SIZES, STD FULL WT. STEEL PIPE
ACTUAL INSIDE DIAMETER OF PIPE IN INCHES
m
WEIGHT OF FLUID, THOUSANDS OF LBS. PER HOUR
M
MASS VELOCITY, THOUSANDS OF POUNDS PER HR. PER SQ. FT. OF CROSS SECTION AREA
VELOCITY IN FT. PER MIN. DIVIDE "M" BY 60 TIMES THE WT. PER CU. FT.
REFERENCE LINE "A"
Δp
PRESSURE DROP LBS. PER SQ. IN. PER FOOT OF PIPE AT 1 ATM.
REFERENCE LINE "B"
Z
VISCOSITY IN CENTIPOISES
TEMP. DEGREES FAHR.
V- STEAM
V- AMMONIA
V- OXYGEN
V- NITROGEN
V- AIR
V- HYDROGEN
R- OXYGEN
R- NITROGEN AND AIR
R- STEAM AND AMMONIA
R- HYDROGEN
F.L. Snyder

change and is almost constant as the radius increases up to a certain point. In bends up to $R / D = 5$ the pressure drop increases. How far beyond this radius the curve rises was not determined. Of course, it is obvious that the larger the radius of the bend, or as the curve approaches a straight line, the lower the pressure drop. At what radius the curve starts downward was not determined because the largest bend available for test was one having a radius of five pipe diameters. When the radius of the bend is smaller than three pipe diameters, the pressure drop is caused by shock which increases turbulence in the connecting pipe immediately following the bend.

Figs. I, II, and III, shown on Plate I portray the above discussion. Fig. I represents a bend with a radius of 1.5 or 5 pipe diameters and shows the turbulent condition of the fluid from the point of curve, to a point in the connecting pipe at which the pressure drop is no longer influenced by the bend. In the case of the bend with a radius of 5 pipe diameters this condition is due to the distance between the points of curve and tangent and is referred to as the time factor of turbulence. The bend with a radius of 1.5 pipe diameters has a more abrupt direction change, causing sufficient shock to increase turbulence in the connecting pipe beyond the point of that in the case of the 5 pipe diameter bend, which balances the time factor of turbulence in the two cases. This applies to Fig. III, which has an extremely abrupt direction change, prolonging the time factor of turbulence in the connecting pipe. Fig. II is formed to a radius which reduces shock, as compared to the 1.5 pipe diameter bend in Fig. I and the time factor of turbulence is of shorter duration as compared to the 5 pipe diameter bend. As the curvature of the bend approaches a straight line the intensity of the turbulence caused in making the direction change is reduced to where only the friction of the fluid on the pipe walls has any bearing on the pressure drop, therefore should be considered as straight pipe, corrected for any irregularities in the pipe wall created by corrosion or erosion.

While no actual tests were made on 45 or 180 deg bends, sufficient data were obtained from which multiplying factors were determined and which can be used with an assurance of accuracy. In the case of the 45 deg bend multiply the pressure loss value of a 90 deg bend by 0.64 and for a 180 deg bend by 1.34.

Contraction and expansion losses, through reducers, into branch lines or receivers of different kinds, may be



determined by the following formulas:

$\Delta p_e = w(u-u_1)^2/144 \times 2g$

$\Delta p_e = Kwu_1^2/144 \times 2g$

$K = 0.4[(1.25 - (a_1/a)]$ when a_1/a is less than 0.715.

$K = 0.75[(1 - (a_1/a)]$ when a_1/a is greater than 0.715.

Chart No. 6 gives the friction multiplication factor f_2 for determining the pressure loss through pipe coils, the equation of which is expressed as:

$$Ap_r = f_2 \Delta p$$

in which:

Δp = Pressure drop in straight pipe, lb per sq in. per ft of pipe as determined by Charts Nos. 1, 2, 3 or 4 or by Poiseuille's formula.

The result of this equation should be multiplied by the roughness factor as given above.

In the absence of any authentic information regarding the pressure drop through gate, globe, angle or check valves, it is suggested that the equivalent length of straight pipe expressed in pipe diameters as shown in the following table be used.

Valves	Pipe Diameters
Gate	5*
Globe	100
Angle globe valves	80
Angle gate valves	30
Check	65
Tilting disc check	25

*Mark's Handbook, 2nd Edition, p. 278.

How to Use Charts Nos. 1 and 2

With water flowing at 500 gpm having a temperature of 150 F, velocity not exceeding 5.57 ft per sec. Determine the size of the pipe and the pressure drop in lb per sq in. per ft of pipe. Refer to Chart No. 1. Convert known values to correspond to those values shown on the chart:

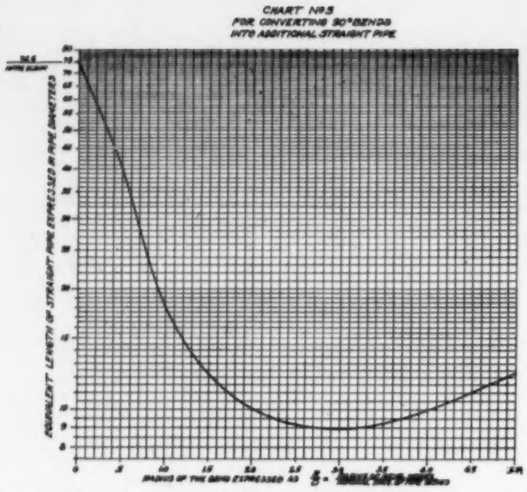
1. The weight of water at 150 F is 61.18 lb per cu ft. Now convert 500 gpm to thousands of lb per hr. 500 gpm = 500 × 60 × 61.18 divided by 7.48 = 245.5 thousands of pounds per hr on m scale.

2. Transpose velocity in ft per sec to Mass Velocity M. $M = 5.57 \times 3600 \times 61.18 = 1227$ thousands of lb per hr per sq ft of pipe transverse section on M scale.

3. Connect 245.5 on m scale with 1227 on M scale along line $X-X$ reading on D scale at point of intersection the nearest nominal pipe size, which in this case is 6 in.

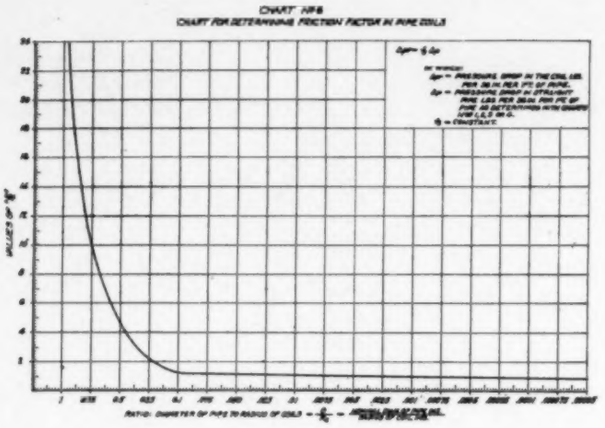
4. Connect 150 F on temperature scale to reference point V-water along line Z-Z, reading on Z scale at the point of intersection the viscosity of the water in centipoises, which is 0.4.

5. Now calculate the Reynolds number m/ZD to determine which chart to use. $m/ZD = 245.5/0.4 \times 6 = 102.25$. This value of Reynolds number 102.25 indicates that the flow occurs in the turbulent flow region where the Reynolds number is greater than 27, which necessitates the use of Chart No. 2. Referring to Chart



Note: To determine the equivalent length of straight pipe in 45 deg bends, multiply the value of a 90 deg bend by 0.64 and by 1.34 for a 180 deg bend.
The radius of a single weld mitre elbow is assumed to be zero.

No. 2, connect the values $m = 245.5$ and $M = 1227$ intersecting D scale and reference line A along line X-X. Connect 150 F on the temperature scale to reference point R-water, intersecting reference line B along the Y-Y line. Connect points X and Y on reference lines A and B intersecting the Δp or pressure drop scale at 0.0075. This value is the pressure drop in lb per sq in.



per ft of pipe and should be multiplied by the roughness factor f_1, which for general use is 1.2. Therefore, the corrected pressure drop will be $0.0075 \times 1.2 = 0.009$ lb per sq in. per ft of pipe.

How to Use Charts Nos. 3 and 4

With steam flowing at 500,000 lb per hr, at an absolute pressure of 250 lb per sq in. and temperature including superheat of 500 F; velocity not to exceed 100 ft per sec. Determine the pipe size and pressure drop in lb per sq in. per ft of pipe. Refer to Chart No. 3.

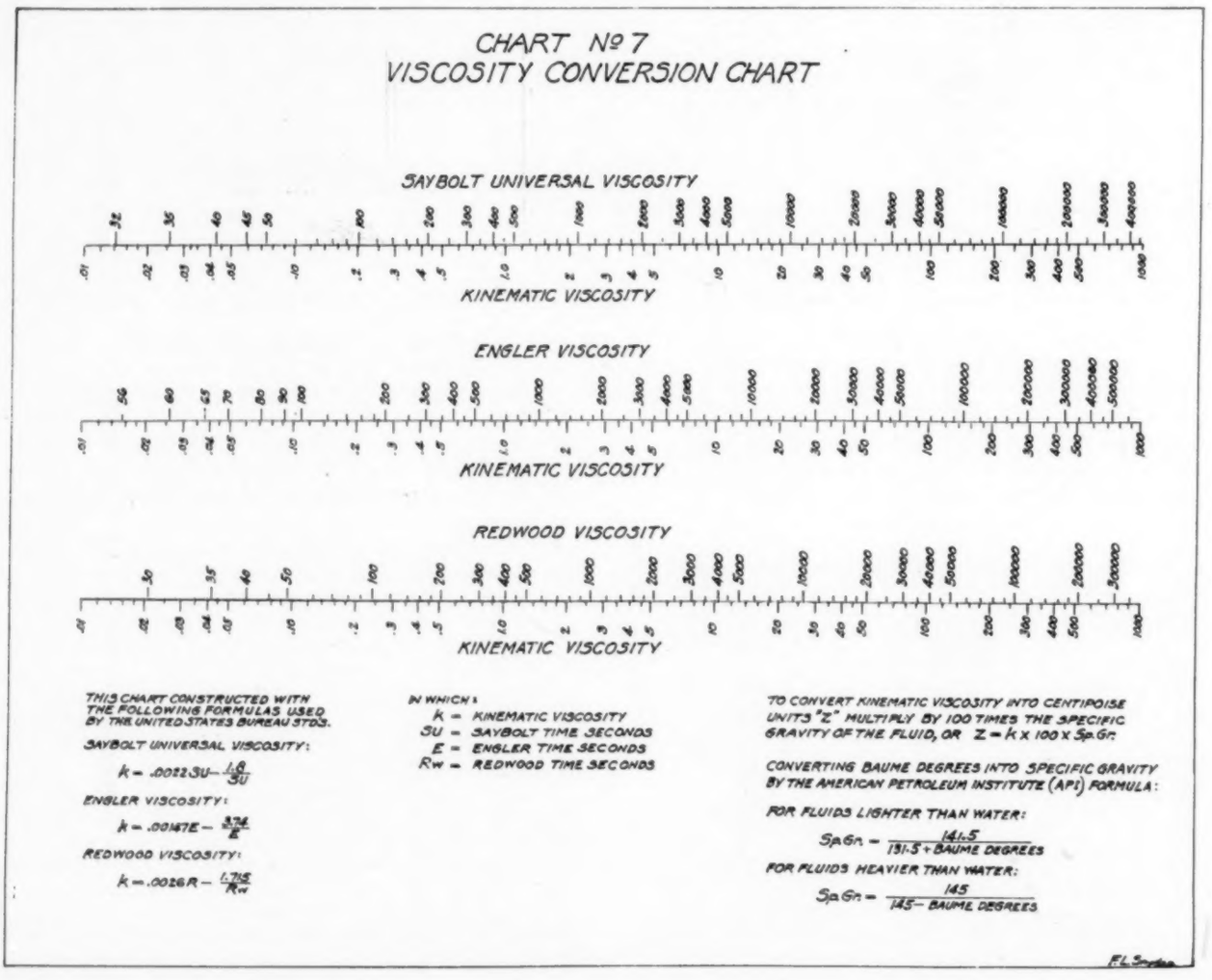


CHART Nº 7
VISCOSITY CONVERSION CHART
SAYBOLT UNIVERSAL VISCOSITY
KINEMATIC VISCOSITY
ENGLER VISCOSITY
KINEMATIC VISCOSITY
REDWOOD VISCOSITY
KINEMATIC VISCOSITY
THIS CHART CONSTRUCTED WITH THE FOLLOWING FORMULAS USED BY THE UNITED STATES BUREAU STDS.
SAYBOLT UNIVERSAL VISCOSITY:
$k = .00223U - \frac{1.8}{SU}$
ENGLER VISCOSITY:
$k = .00147E - \frac{3.74}{E}$
REDWOOD VISCOSITY:
$k = .0026R - \frac{1.715}{Rw}$
IN WHICH:
k = KINEMATIC VISCOSITY
SU = SAYBOLT TIME SECONDS
E = ENGLER TIME SECONDS
Rw = REDWOOD TIME SECONDS
TO CONVERT KINEMATIC VISCOSITY INTO CENTIPOISE UNITS "Z" MULTIPLY BY 100 TIMES THE SPECIFIC GRAVITY OF THE FLUID, OR $Z = k \times 100 \times Sp.Gr.$
CONVERTING BAUME DEGREES INTO SPECIFIC GRAVITY BY THE AMERICAN PETROLEUM INSTITUTE (API) FORMULA:
FOR FLUIDS LIGHTER THAN WATER:
$Sp.Gr. = \frac{141.5}{131.5 + BAUME\ DEGREES}$
FOR FLUIDS HEAVIER THAN WATER:
$Sp.Gr. = \frac{145}{145 - BAUME\ DEGREES}$

Convert known values to correspond to values on the chart:

1. The weight of the steam being given is referred to as 500 thousands of lb per hr on m scale.

2. Transpose velocity in ft per sec to Mass Velocity M. The weight of the steam in this case is 0.496 lb per cu ft, then $M = 100 \times 3600 \times 0.496 = 178.5$ thousands of lb on M scale.

3. Connect 500 on m scale with 178.5 on M scale along line X-X reading on D scale at the point of intersection the nearest nominal pipe size, which in this case is 24 in.

4. Connect 500 F on the temperature scale to reference point V-steam along line Z-Z, reading on Z scale at the point of intersection the viscosity of the steam in centipoises, which is 0.0181.

5. Now calculate the Reynolds number m/ZD to determine which chart to use. The inside diameter of a 24 in. pipe used for 250 lb steam is 22.626 in., then $m/ZD = 500/0.0181 \times 22.626 = 1151$. This value of Reynolds number 1151, indicates that the flow occurs in the turbulent region where m/ZD is greater than 27. This necessitates the use of Chart No. 4. Refer to Chart No. 4. Connect the values $m = 500$ and $M = 178.5$ intersecting D scale and reference line A along line X-X. Connect 500 F on the temperature scale to reference point R-steam, intersecting reference line B along the Y-Y line. Connect points X and Y on reference lines A and B intersecting the Δp or pressure drop scale at 0.115. This is the pressure drop in lb per sq in. per ft of pipe at atmospheric pressure and should be multiplied by the roughness multiplication factor f_1, which for general use is 1.2. Since the pressure drop is given in lb per sq in. at atmospheric pressure and as the pressure of the fluid increases, the pressure drop decreases, it is necessary to correct it for the higher pressure, which in this case is 250 lb per sq in. absolute. The same equation may be used in making the roughness factor correction. Then, the final pressure drop equals atmospheric pressure multiplied by the pressure drop times the roughness factor divided by the working pressure, or $14.7 \times 0.115 \times 1.2/250 = 0.00812$ lb per sq in. per ft of pipe. If gage pressure were used then the equation would read $14.7 \times 0.115 \times 1.2$ divided by $14.7 + 250$.

Chart No. 5 is used to convert 90 deg bends into additional straight pipe and arranged to relate its use with Charts Nos. 1, 2, 3, and 4. The pressure drop in a 90 deg bend is obviously greater than that in straight pipe of equal linear length and varies with the radius of the bend. Although this loss is not constant in different pipe sizes and bend radii, it is so nearly so that this chart is plotted along the line of mean average. As an example wherein the use of the chart will simplify pipe design, consider the pressure drop expressed in additional straight pipe through a 6 in. dia. 90 deg bend having a radius of 1.5 pipe diameters. Referring to the $1.5 = R/D$ line on the chart and following this line vertically to the point of intersection on the curve, then straight across to the left, read 12.3 pipe diameters; this value multiplied by the diameter of the pipe, divided by 12 or $12.3 \times 6 / 12 = 6.15$ ft of additional straight pipe. If a pipe line has 50 ft of straight pipe and one 90 deg bend having a radius $R / D = 1.5$ and the pressure drop is 0.009 lb per sq in. per ft of straight pipe (see above) then the total drop including the bend would be $(50 + 6.15) \times 0.009 = 0.5054$ lb per sq in. If the pressure drop expressed in additional straight pipe for a 6 in. dia. 90 deg bend is 6.15 ft, then for a 6 in. dia 45 deg bend having the same radius multiply 6.15 by 0.64 and for a 180 deg bend multiply by 1.34.

How to Use Chart No. 6

Find the pressure drop in lb per sq in. per ft of pipe in a pipe coil having a radius of 60 in. and fabricated from 6 in. dia. pipe in which 500 gpm of water flows at a temperature of 150 F. From above, it is found that the pressure drop in a 6 in dia. straight pipe with water flowing at 500 gpm is 0.009 lb per sq in. per ft of pipe. Now calculate the ratio of pipe diameter to the radius of the coil. In this case the ratio $6 / 60 = 0.1$. Referring to the 0.1 line on the chart and following this line vertically to the point of intersection on the curve and then straight across to the left read 1.3, which is the value of the constant f_2. Now calculate the pressure drop in the coil with the equation $\Delta p_r = f_2 \Delta p$ or $\Delta p_r = 1.3 \times 0.009 = 0.0117$ lb per sq in. per ft of pipe.

Chilled Water Pump Installation for Steam Jet Jobs Requires Care

By **J. J. Hayes***

ONE application of pumps to air conditioning which requires care in installation for satisfactory operation is the removal of chilled water from the evaporator chamber of a steam jet refrigerating system. The chilled water pump on a system of this kind is removing water from a chamber under high vacuum—29.75 in. for 40 F water, for instance. The water would flash in the rotor of the pump because of lowering of pressure by the pump suction if the proper practices were not observed.

The water line in the evaporator or flash tank must be a certain height above the pump inlet, usually from 4 to 5 ft. The piping should be of ample size and as straight as possible to reduce friction loss to the minimum—in other words, the pump should be given as favorable conditions as possible to do its job. The pump should have a water-sealed stuffing box, equalizing connections from the eye of the rotor, and from the top of the volute back to the evaporator tank to vent any air liberated in the pump. A needle valve in these lines permits adjustment.

Trouble with a recent steam jet installation stressed the importance of these precautions. The chilled water pump failed to work properly. It was found that the suction piping had a number of unnecessary elbows, and that there was no equalizing connection. The existing pump—which would probably have worked satisfactorily if it had been installed correctly—was replaced with one having tight shaft seals, the suction piping was made larger and straighter, and an equalizing connection as described was installed, all of which corrected the difficulty.

*Stannard Power Equipment Company, Chicago.

AIR CONDITIONING RESCUES INDUSTRY FROM THE WEATHER . . .

THE Waverly Press, Baltimore, Md., nationally known printers of fine technical books and medical literature, realized the desirability of year-'round humidity control in their plant. Winter control of humidity had been provided for several years, but difficulty in producing precision printing was still experienced in summer. The condition of the paper, which changed with changes of humidity, had some bearing on the problem, but the principal difficulty was caused by variation in thickness of the cherry-wood blocks on which the metal plates are mounted prior to their being placed in the presses. These wood blocks, because of their hygroscopic nature, contract and expand with changes of atmospheric conditions, making it extremely difficult to maintain the precision desired.

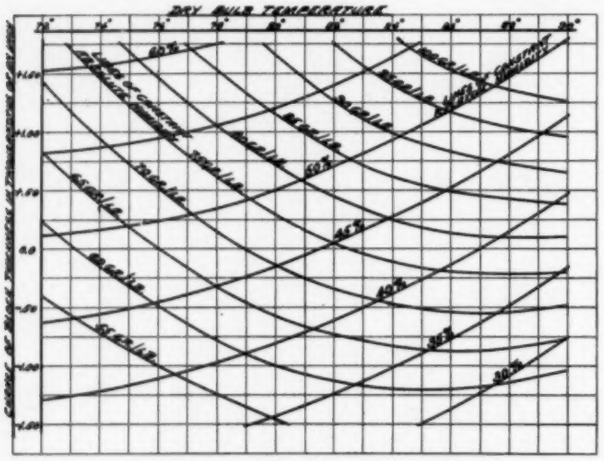


Fig. 1—Results of investigation to determine effect of varying air temperatures and humidities on the thickness of the wood blocks on which the printing plates are mounted. A special instrument was developed in order to take these readings

Behavior of Wood Blocks Studied

Before designing the humidity control system, it was necessary to study the exact behavior of the wood blocks with varying air temperatures and humidities. These tests, naturally had to cover a period long enough to include all seasonal changes in weather and necessitated the use of equipment not available in general test work.

H. H. Tunis, Research Engineer of the Press, devised an instrument which for want of an established name was called a "Hygro-Xylometer."[1] In order to check the effect of the air on the blocks, hygro-xylometer readings were taken regularly in the composing room and in the press room. The instrument consisted of a cherry-wood block so connected to a dial indicator that variations in a wood block 0.853 in. thick could be measured in hundred-thousandths of an inch. The results of these investigations are plotted in Fig. 1. In plotting these final results, it was necessary to take into consideration the time of exposure, as it was determined that the time element was a factor in changes of block thickness. Occasionally, there was a lag of many hours before the blocks came to their equilibrium thickness in response to a change in humidity or temperature.

An inspection of the curves in Fig. 1 discloses the rather interesting result that the absolute humidity must be increased slightly as the temperature increases to secure uniform block thickness. In other words, when the temperature increases, the consequent drop in relative humidity tends to dehydrate and shrink the wood. On the other hand, constant relative humidity with increasing temperature tends to increase the size of the blocks. Consequently, the condition required for constant block thickness is a compromise between constant absolute and constant relative humidities with changes in temperature.

Adsorption Type of Air Conditioning System

The next step was the selection of the system of humidity control to produce the conditions required. Calculations were made and estimates were prepared using several methods of producing the desired conditions. The adsorption method is well suited to humidity control and the comparison of the several methods indicated

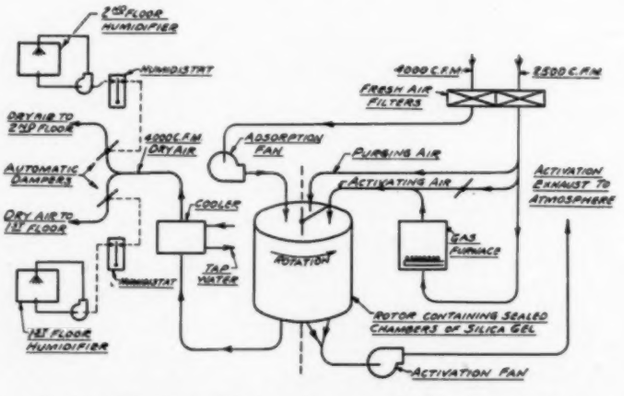


Fig. 2—Schematic diagram showing air flow through the air conditioning equipment. The various items shown are in approximately their respective positions, as illustrated in the view of the completed installation, Fig. 4

[1]From the Greek "Hygros" meaning "water" and prefix "Xylo" meaning "wood." The meaning is therefore a meter for measuring the water content of the wood.

that this was the most economical to employ for this particular job.

The composing room, as well as the press room, is air conditioned, because the wooden blocks must be seasoned for many days before use. The blocks must be kept in the same atmospheric condition during this period as when they are in storage or in use on the presses.

The total volume of the conditioned spaces is 134,000 cu ft. Humidity control is provided by a silica gel dehumidifier which treats 4,000 cfm of fresh air. This equipment removes approximately 65 per cent of the moisture from this air, after which the air is cooled and delivered to the various conditioned spaces. Cooling of the dehydrated air is accomplished by indirect surface coolers, using city water at a temperature of 72 F. No further cooling is required, but a total of 15,000 cfm of air is recirculated locally in the rooms to secure the desired ventilation. Since the thermal capacity of the building is large, the inside temperature rarely exceeds 88 F regardless of outside extremes. An absolute humidity of 72 grains per pound of dry air is provided, and this corresponds approximately to a relative humidity of 40 per cent at a temperature of 85 F.

. . . . . *By* **Bayard P. Fonda***

Air conditioning has come to the rescue of many an industrial process, preventing lost time, spoilage of the product, high production costs. The problem of variable humidities in printing plants — as an example — has been grappled with for years, as it means lost time, imperfect impressions, sometimes a shut-down.....This article describes an unusual air conditioning system installed by The Waverly Press for control of air conditions in summer as well as in winter

Dehumidifier of Novel Design

The dehumidifier, of novel design, is the continuous rotary type. Reactivation of the silica gel takes place in one sector of the rotor, through which a stream of heated air is drawn by the activation fan. The effluent moisture-laden air is discharged to atmosphere. In another sector of the rotor, simultaneously with the above operation, fresh air is being dehydrated by the silica gel. Subsequently, this air is cooled by tap water (as explained before) and fed into two main dry air supply ducts. The rotation of the adsorber or container for the silica gel beds which has a circumferential drive, continuously feeds beds of reactivated gel into the adsorbing sector and saturated gel into the reactivating sector. This results in continuous operation when and as needed. Fig. 2 is a schematic diagram of the air flow through the equipment. Fig. 3 shows the dehumidifier completely constructed in the shop before it was installed, and Fig. 4 shows the completed installation.

To the right in Fig. 4, is seen the gas-fired activation furnace which heats the air to the temperature necessary to drive the moisture from the saturated silica gel. In the center is the slowly rotating gel bed container, with the fresh air ducts and activation air ducts above it. The dried air leaves the container from the lower side and is conducted to the upper left where it passes through the indirect cooler, already described. Just below the cooler and mounted on the wall are the damper motors which govern the operation of the distribution dampers.

The maximum hourly operating quantities which would be required for continuous operation of the dehydrator are 900 cu ft of gas at 500 Btu per cu ft, 4

(*Continued on page* 16)

*The author was air conditioning consultant for the Waverly Press at the time this installation was made and designed the unit herein described; he is now connected with The Bryant Heater Company as air conditioning engineer.

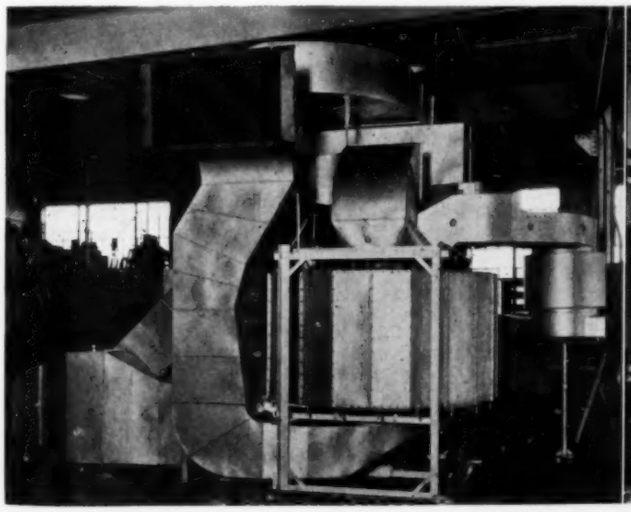


Fig. 3—The dehumidifier, completely constructed in the shop before it was installed on the job

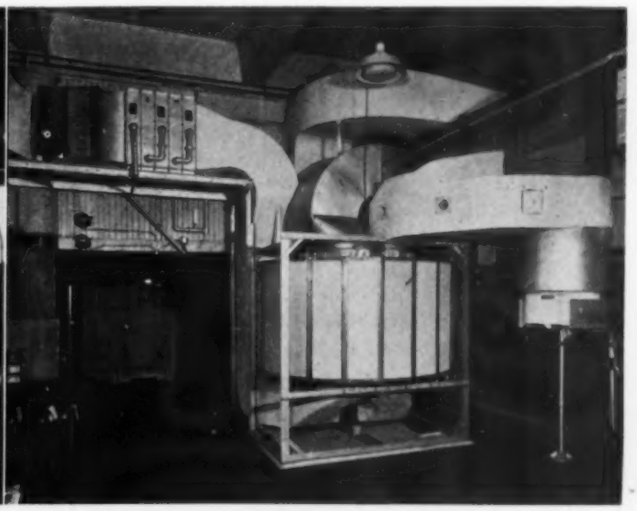


Fig. 4—The dehumidifier installed at the plant of The Waverly Press.

Consulting Engineer Tells How to Improve

THEATER AIR CONDITIONING

By **D. D. Kimball***

DURING thirty-five years of theater ventilation work the writer has seen remarkable changes in theater design, construction and equipment—from the theater having orchestra, balcony and gallery, with many posts or hangers (the gallery having a terrific pitch and benches), to the theater having a huge orchestra floor and great balcony, without columns; and from little or no ventilation to the modern cooling installation.

In 1900 when the writer designed an auditorium ventilating installation with the air supply of 30 cfm per person introduced through the ceiling, with but slightly less exhaust at the floor, both supply and exhaust openings being uniformly distributed, precedent for a downflow system was demanded by the owner (and found). When in 1913 the writer published a paper recommending the use of recirculated air in ventilation the idea was ridiculed. Today it is in general use and cooling installations would be next to impracticable without it.

Twenty years ago a theater provided with a supply of air equal to 10 cfm per person was thought to be well ventilated in spite of discomfort experienced by the patrons. Then, the air was almost universally introduced through the floor and exhausted through the ceiling or rear walls. Chilled legs, dusty air, and uneven temperatures were the results. Today a bountiful supply of clean, conditioned air from the ceiling, from which level it passes down through the theater picking up the heat and moisture given off by the patrons but passing over them at the desired temperature and then being drawn out through the floor, is the ideal to be attained—though many installations fall short of the goal.

As to the types of systems found in this survey[1] those theaters having no ventilation were very old buildings (one of them eighty-four years). Those having supply or exhaust only were also old buildings. Those having reversible systems as well as those listed last month as "nondescript" were largely neighborhood houses, usually small and cheaply built and representing aggressive selling of equipment, rather than engineering. Those theaters in which the air was supplied through the floor and exhausted through the ceiling (or in some cases through the rear walls) were older buildings. In all of the newer buildings air was supplied through the ceiling and exhausted or returned through the floor (orchestra floor, and usually balcony floor also), thus showing a complete reversal from the practice of earlier years. It is consequently amusing to think of the difficulty experienced in inducing an owner to accept such a design only fifteen years ago.

Distribution of Air: To the limited—very limited—and unequal distribution of the air moved by the ventilating system through the theater may failures of some systems be directly attributed. Lack of air distribution is almost always found to be due to uneven distribution of return air outlets. Reasonably perfect general distribution of the return air outlets is seldom found. Although twenty-six houses were classed as having "generally distributed outlets," in the majority of cases such outlets were located at aisle seats only and while this is better than the average method it is far from ideal. In many cases the return air outlets were found exclusively at the rear, front or sides of the house, with resulting high temperatures at points remote from these outlets and temperature differences of as much as 8 F on the same floor. No such difficulties are found where the return air outlets as well as the supply inlets are very generally distributed. Balconies and galleries without direct return air outlets are higher in temperature and less comfortable than are those areas of the theater having return air outlets.

Accessory Spaces: What little sympathy with or respect for the ventilation of theater toilets by means of windows the writer may ever have had has completely vanished as a result of this survey; this applies as well to the use of windows for ventilating toilets in theaters having the so-called "plenum" system of ventilation. The air passing through the window is just as likely to be blowing in as out, carrying the odors into the theater, frequently the case even with the "plenum" system.

Where no windows occur in the toilets and gravity vent ducts are provided therefrom, dependence being placed on the plenum, ventilation is inadequate or *nil*. Nor will 8 to 10 changes of air per hour provide adequate toilet room ventilation. After years of use and with the best of "housekeeping," these rooms are bound to be, and are found to be, odorous unless amply, positively and directly ventilated with an independent exhaust fan system. Disc or propeller type fans placed in the windows or walls do not assure satisfactory results inasmuch as an opposing wind pressure will often nullify the work of the fan.

Means of air inlet to toilets is essential to their ventilation. For smaller toilets this may be by means of grilles in the door or by cutting off the bottom of the door, while in the larger toilets an air supply directly or through the adjacent lounge is desirable. If a direct toilet room air supply is provided it should be slightly less than the amount of the exhaust to prevent any movement of air from the toilet to adjacent rooms.

*Consulting Engineer, New York City. Member of Board of Consulting and Contributing Editors.

[1]This article is based on a recently completed survey of seventy-six theaters as described by the author in the December, 1934, H. P. & A. C.

Large lounges always should be provided with a direct air supply and exhaust. While some of the air from this room should be exhausted through the adjacent toilet only rarely is it wise to thus arrange all of the exhaust from the lounge.

The foyers are usually so integrally a part of the theater proper that their ventilation is and should be a part of the main theater system.

In large theaters the best provision for combating outside air pressures to prevent drafts of air into the theater from the lobby is that which comprises a separate fan drawing return air from the return air chamber under the theater and discharging it into the foyer across the doors or parallel to the sill of both outer and inner doors. Properly arranged, this will prevent hot outdoor air from reaching the orchestra floor in the summer, and with auxiliary indirect radiation it will heat the foyer in winter and prevent cold air from reaching the orchestra or foyer.

Thirty-five years of experience in the air conditioning and ventilation of theaters and a recently completed survey of the mechanical equipment in seventy-six motion picture houses is the background for this informative article, which discusses—among other things—the importance of air distribution, the need and opportunity for theater comfort cooling, importance of air filters in reducing cleaning costs, possibility of using well water with refrigeration to cut cooling costs. The writer's experience gives authority to his views; his survey of actual conditions, described last month, assures their practicality.

Advertising at the outer doors to call the attention of passers-by to the cooling installation is a cheaper method of informing the public about the cooling of the theater than is the forcing of large quantities of expensively treated air out to the street.

Exhaust fan ventilation is universally provided for the projection machines, and generally another fan exhaust system is provided for the projection room itself, as should be the case. Many projection rooms are very inadequately ventilated, for the output of heat therein is very large. Where outside windows do not exist in such rooms a direct supply of air is desirable, which should not be discharged into the strata of heated air at the ceiling, but thoroughly diffused into the room at a lower level.

Cooling Installations: The fact that less than half of the theaters visited were provided with cooling systems indicates that there is still a considerable opportunity for the installation of such plants.

In every instance of a cooling installation the air was supplied through the main ceiling and through the soffits of the balcony and gallery (where these existed). In the matter of the return air the practice was less uniform, as described above. It was mostly on this latter basis that a number of these installations failed to be classed by the writer as "good," because the result of the inadequate general distribution of the return outlets could be readily detected in the atmospheric conditions of these theaters.

In a few cases amazing things were found to have been sold as "cooling systems."

Dehumidifiers: Dehumidifiers with two and with three ranges of spray nozzles and with or without direct expansion coils within the casing, were found to give very similar results. It is suggested that spray nozzles should be attached to the riser pipes by means of brass nipples and that the bottom cross headers should pitch in the direction of the water flow, these to be provided with a large size drain valve or plug at the far end so that these pipes may be readily washed out.

Air Filters: In but two instances were air filters found, but plenty of evidence—such as ceilings and decorations badly soiled about the air inlets—indicated that all air entering the theater through the ventilating system should be filtered. One theater manager complained that he was maintaining a dry cleaning establishment because of having so often to pay for the cleaning of patron's dresses which were soiled by dirt dropping from the ceiling.

Generally a considerable amount of air is bypassed around the dehumidifier and is therefore not washed at any time; also during the spring and fall there are several weeks during which there is no air washing and no air conditioning. The result is that with the air blown in close to the ceilings and on the same plane, many ceilings are badly soiled and the cost of cleaning (even where possible) is considerable.

The cleaning of many of the air filters heretofore installed is troublesome, consuming much time and labor, with the result that some of the filters which have been installed have been abandoned. None the less it is important that with ceiling outlets of the usual pan type all of the air blown into the theater should be filtered. The air filter should be of a type easily and inexpensively cleaned. A type of air filter which may be easily cleaned by vacuum cleaning tools is often desirable inasmuch as practically all theaters have vacuum cleaning systems.

Air Heaters: Aside from a few old type pipe coil blast heaters found in old buildings, all air heaters were either cast iron vento or aerofin, both of which seemed to give about the same degree of satisfaction.

Duct Systems: The classification of duct systems as "ample" or "deficient" or "well installed" or "poorly installed" (see article last month) was purely arbitrary.

Where an insufficient amount of duct work was found the greatest was usually in the return air duct work. The major fault found was short bends and connections, lack of control dampers, and leaky construction due to poorly made seams and allowing trusses, beams and ceiling hangers to pass through ducts; also poor connections to ceiling outlets. The leakage is sometimes so great that a noticeable plenum is found in the attic space and much air intended for the auditorium is lost. Asbestos cement or plaster-of-paris is a poor substitute for well constructed duct work.

Some duct systems were unnecessarily complicated and not skillfully designed, with the ducts badly placed. Very few insulated ducts were found.

Refrigerating Machinery: Generally the refrigerating equipment was adequate, well arranged and nicely installed. A few cases of limited capacity and of undesirable crowding of the machinery were noted; also a very few cases of operating noise which might be detected in the theater were observed. The sound of the water passing through the piping from the machinery in the basement up to the dehumidifier in the attic was detected in the orchestra of two theaters. Insufficient bracing of CO_2 lines was observed in a few cases.

As for a preference for the use of a single unit or two units together to provide the desired capacity, two units seem desirable in the larger theaters, but if one unit is out of commission during very warm weather the remaining unit serves but little purpose.

Wells: The utilization of deep well water in air conditioning systems is just beginning to receive its proper due. Some assume that the use of well water will eliminate altogether the need of any refrigeration for comfort or theater cooling and this is a serious mistake. An attempt to use well water without operating the refrigerating machine in one case caused water to run from the ducts and loaded the theater with moisture. Well water will rarely be found cold enough properly to dehumidify a theater, but where an ample supply of well water can be had it will materially reduce the size of the refrigerating machine required, reduce the cost of the installation and largely reduce the operating costs.

This well water may generally be used for precooling and then serve as condensing water.[2]

In some districts it is required that this water be returned into the ground through a return well. Double casings in a single well have been used for this purpose, the water being drawn from a deep level and returned at a higher level.

[The author will continue his discussion of air distribution in a third article.—THE EDITOR.]

[2]See August, 1934. H. P. & A. C., pp. 344-345 and October, 1934, pp. 431-432.

Waverly Press—

(*Continued from page* 13)

kw-hr of power, and 240 cu ft of 72 F water. It has been found that under maximum outside conditions, the dehydrator operates approximately three-quarters of the time.

Control of the System Presented a Problem

The selection of a hygrostatic control presented quite a problem as it was necessary to maintain an arbitrary atmospheric condition—a condition independent of temperature and independent of humidity, as is indicated on the zero deviation line of Fig. 1. A control system was finally devised to meet the requirements:—

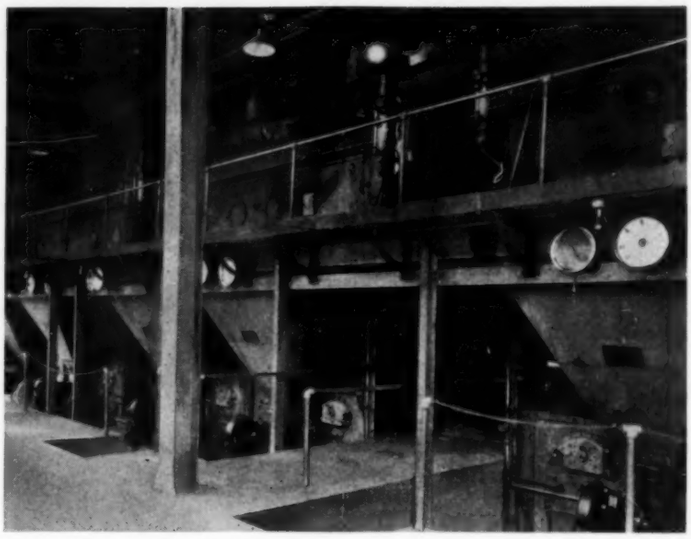
Each of the two floors which are conditioned has a main dry air supply duct equipped with an automatic damper. A human hair humidistat on each floor operates a damper motor to shut off the supply of dry air when the humidity drops to the humidistat setting. In the event that both humidistats are satisfied, the two dampers automatically open to furnish a supply of fresh air and the dehumidifier automatically shuts down. These humidistats, however, are really secondary or actuating controls. The primary control consists of a series of wooden blocks, identical to those used on the presses, so arranged as to magnify greatly any deviation in block thickness. The motion resulting from such deviation is transferred to the adjusting arm of the humidistat, thus shifting the working range upward or downward as required. The "high" contacts on the humidistat open the dry air dampers, and the "low" contacts turn on the sprays in the several humidifier units which are installed. There is, of course, a normal or neutral floating point, where neither the humidifiers nor the dehydrator are called upon to operate. Thus, with this arrangement it is the blocks themselves which dictate the proper atmospheric condition for maintaining constant block thickness.

This air conditioning system was put in operation in the spring of 1934 and the results have been extremely gratifying. The following is quoted from an article on the application of this system to the precision method of printing, appearing in a recent issue of The Waverly Press house organ:

"The result of this system is demonstrated by the study of one month under controlled atmosphere. About six hundred cuts were printed and not one was removed from press because of block thickness. With uncontrolled atmosphere, over 25 per cent blocked cuts had to be removed and corrected for type height."

"FLORIDA FLYER" OF THE EASTERN AIR LINES, which went into service last month, has thermostatically controlled steam heating and individually controlled ventilation for each passenger. Air is taken in at the nose and distributed by ducts to the passenger space, the outlet at each seat being under the control of the occupant

HOW IMPROVE HEATING?



Photo, Detroit Stoker Co.

"Savings in operation are . . . likely if . . . coal is fired mechanically." When the boiler plant at The Homestead, Hot Springs, Va., was modernized with the stokers shown here savings of 90c per ton in the cost of coal resulted—and 10,000 tons of coal are required annually. In addition, ability to maintain steam pressure with fluctuating demands, and elimination of smoke, are "stoker dividends"

By **Samuel R. Lewis***

HOW to improve the heating of a plant or building and reduce heating cost is a question which can be answered in many ways, depending upon the particular situation in each specific case. Sometimes it involves a change in the method of steam generation, the fuel or the way it is fired, distribution of the steam to the heat-emitting surfaces, the heat-emitting surfaces and their accessories themselves (for instance, installing unit heaters in place of pipe coils and replacement of traps, etc.), temperature and zone control, etc. One question that frequently comes up is the advantage of changing from high pressure to low pressure steam, and even here the answer is not always the same. Frequently the change is justified; in other cases, a survey indicates otherwise.

The case under consideration is an example of the latter. The question of whether or not a change from high to low pressure steam for heating, cooking, water heating in a hotel would be advisable has been asked by a reader of H. P. & A. C., an operating engineer. From the information supplied, it appears that the installation of stokers and some form of temperature control would be the better way of securing the desired economy. The reasoning underlying these recommendations is given here, principally to indicate what must be considered in arriving at a decision of this kind, and should not be construed as a "blanket recommendation." Each case needs individual study, the answer depending—as mentioned—on the situation.

Briefly, the plant under consideration has three hand-fired down-draft boilers. It heats the hotel and furnishes steam to the food cooking apparatus in the kitchen at 20 lb pressure and steam for hot water heating at 15 lb. The boilers are operated at 75 lb normal steam pressure, and there are two high pressure steam lines running about 250 ft from the boilers, where reducing valves are placed to deliver steam at about 3 lb to the heating system, which has modern thermostatic return traps on each radiator. The following questions are asked, to each of which the writer's replies are appended:

1. *If the boiler pressure were reduced to 4 lb, what size low pressure steam main should replace each present high pressure main leading to the reducing valve on the heating system, if each present high pressure main supplies 10,500 equivalent square feet of direct radiation?*

*Consulting Engineer, Chicago, and Member of Board of Consulting and Contributing Editors.

A QUESTION AND THE ANSWER

The Editor: I would greatly appreciate any information on the following matter:

A change of management has recently taken place in the hotel building of which I am an operating engineer. In this plant we carry *high pressure* steam (75 lb gage) for heating the building and for process work. The new management feels that by changing the plant to *low pressure*, we will economize in fuel as well as labor and that the cost of making the necessary changes will be paid for by these savings in a very short time.

Following are some details of the plant and equipment. Total radiation amounts to about 37,000 sq ft; boilers are down draft of about 105 hp each; two of these boilers carry the peak load in winter and only in severely cold weather do we have to cut in the third boiler. This has happened only three times in the twelve years that this plant has been in operation. Our efficiency, according to an evaporation test that we made last year, showed 62½ per cent, which is unusually good for hand-fired boilers.

For heating purposes we reduce our pressure to 3 and 4 lb. The reducing valves are located about 250 ft from the boilers, thus making a total of about 500 ft of mains. One main is 3 in. in diameter and the other 4 in.

For process steam we reduce our pressure to 20 lb; this takes care of two stock kettles, two steam tables, four coffee urns, etc.

For domestic water we reduce our pressure to 15 lb and this takes care of three tanks equipped with submerged heating coils; the three tanks have a total capacity of 3,400 gallons of water.

The following are my questions:

1. To what size should the mains be increased, if each is to take care of about 10,500 ft of radiation?
2. What changes are necessary in the process equipment?
3. Do the hot water heaters need any alterations?
4. Does it take less fuel to operate at low pressure than at high pressure? If so, why?
5. In general, where are the possibilities of saving, outside of discharging the present licensed men and replacing them with cheaper labor?

The longest run from the boiler to the most distant heating unit along the piping, including an allowance for the friction due to fittings, probably may safely be estimated at 800 ft. The absorption of pressure loss by friction in the piping may be taken as half the initial pressure. Then $4 \div 2 = 2$ lb available, and the pressure drop per 100 ft of run would be $(2 \times 100) \div 800 = \frac{1}{4}$ lb.

A 5 in. pipe will supply 9,100 equivalent sq ft of radiation at a pressure drop of ¼ lb per 100 ft run. A 6 in. pipe will supply 14,900 sq ft at a ¼ lb drop per 100 ft run.

With covered mains, well drained, it is apparent that a 5 in. main could serve sufficiently well. The present 3 in. high pressure main would have, under these changed conditions, a capacity of less than one-third the required value.

2. *What changes are necessary in the process equipment?*

It is probable that the steam cooking apparatus has very small pipes and valves, and very small pipe tappings. While the pressure is reduced from 75 lb to 20 lb, it is probable that the cooking apparatus is not far-distant from the boiler, and that therefore the temperature of the steam which enters the jackets of the cookers is close to 320 F rather than at the approximate 260 F corresponding to the 20 lb actual pressure. This heat was in the steam at 75 lb and remains as superheat for some time after the pressure is reduced through a reducing valve.

It is very doubtful whether a temperature 60 F lower than that at present would serve for the cooking, and it is probable that continual complaint and dis-satisfaction would be experienced with so low a temperature. Eventually gas would have to be used in the cooking apparatus at a considerable cost for the physical changes and at a relatively high cost compared to coal for the gas burned.

3. *Do the hot water heaters need any alterations?*

These heaters, like the kitchen equipment, have been receiving superheated steam and they will have a very much reduced output when saturated steam is substituted. It is not safe to say whether or not they would serve sufficiently well under the changed conditions, but it is probable that they would not have to be changed, if their location is close to the boilers.

4. *Does it take less fuel to operate at low pressure than at high pressure?*

Theoretically, to deliver any given amount of steam at 5 lb pressure requires not only heating the water to 228 F and the input of 962 Btu per lb after that to change the water to steam without increasing its temperature, but requires also a definite addition of heat for every temperature increase. Thus at 4 lb pressure a pound of steam contains about 1,156 Btu, while at 75 lb pressure, it contains about 1,186 Btu.

Expressed in another way, if you have 40,000 sq ft equivalent direct radiation which condenses ¼ lb of steam per hr per Edr, the 10,000 lb of steam required per hr if delivered at 75 lb pressure will have 1186 — 1156 = 30 Btu per lb × 10,000 lb = 300,000 Btu beyond that necessary to fulfill the requirements of the output design of the ordinary heating system. At an efficiency of 62½ per cent, however, this means, with 10,000 Btu coal, $\frac{300{,}000}{10{,}000 \times 0.625} = 48$ lb of coal per hr.

This amount of heat will be required, not for every hour of the heating season, but only during the hours of maximum demand. To establish a seasonal fuel rate one method is to divide by 3 the product of the maximum fuel demand per hour and the total hours. If there are 5000 hours per heating season, then $\frac{5000 \text{ hr} \times 48 \text{ lb}}{3}$ would give some idea of the excess coal per year due to high temperature steam through the heating season; 80,000 lb or 40 tons.

The heat equivalent to this 48 lb of coal per hr, when its use in the building is concerned, is expressed as superheat in the reduced pressure steam. Most of it will be available in the heat transmitting units, because the superheat will bring about the delivery of "all steam" and not steam mixed with water at the remote radiators. Therefore it is not fair to assume that anything like the full value of 40 tons of coal per season would be saved in this plant by changing to low pressure generation.

It is very difficult, with hand firing, to maintain a uniform low steam pressure such as 5 lb; due to fluctuations in demand and in combustion rate, without any built-up reserve such as now is present in the high pressure steam. There is evidence that losses in efficiency of combustion at low pressure, burning coal manually, will far exceed the losses due to warmer gases going up the chimney caused by hotter steam at the higher pressure. There is evidence that more coal actually was burned with manual firing after changing to low steam pressure than was burned when operating at the higher pressure and then reducing the pressure to suit the individual service in a number of cases.

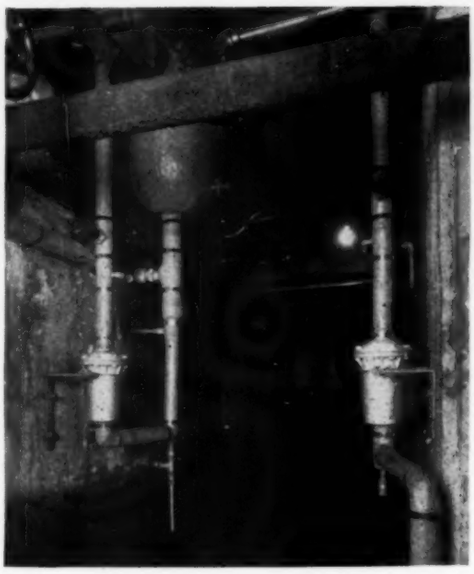
5. *In general where are the possibilities of saving, outside of discharging the present licensed men and replacing them with cheaper labor?*

There is no actual saving likely, except as it may be achieved by reducing wages paid to labor. It is doubtful whether any well managed hotel could put up with a chief engineer, whether operating at low pressure or at high pressure, of less calibre and attainments than would be acquired at the rate of pay of a licensed engineer.

In general, savings in operation are far more likely to accrue if, at the steam pressure as at present, the coal is fired mechanically as in a stoker. There are almost invariably such savings possible in this direction that the cost of stokers will be saved in two or three years.

There are very remarkable savings possible by installing zoned automatic temperature control so that by operation of valves on the steam supply risers controlled by central thermostats, for instance, overheating and wasteful opening of windows will be reduced. These arrangements are being installed in a large number of buildings, both old and new, and in many instances are repaying their cost in savings inside of three years; thereafter rendering a substantial profit.

The writer's recommendation, therefore, based on the information given and without having visited the hotel in question, would be not to change to low pressure operation.



Trap draining a large receiver separator

SELECTING TRAPS for STEAM PIPING

By T. H. Rea*

. . . . . is explained here, supplemented by the necessary tables and examples. Traps on steam distribution mains must do two things—remove water of condensation without steam loss, and air or other incondensable gases. Condensation due to warming up a cold line as well as from radiation loss must be considered, and the traps must be installed so that the condensation will get to them

TRAPS on steam distribution piping must do two things—*first,* remove water of condensation as fast as formed without steam loss, and *second,* remove the air or other incondensable gases mixed with the steam or present in the line before the steam is turned on.

Warming Up a Cold Pipe Line

When steam is first admitted to a cold pipe line, considerable steam is condensed in bringing the pipe up to approximately steam temperature. This initial flood of condensate may be the governing factor in trap selection. Hence it is necessary to calculate this load, using the simple formula:

$$C = \frac{W \times (t_1 - t_2) \times 0.114}{L}$$

where

C = Amount of condensate, lb
W = Total weight of pipe, lb
t_1 = Final temperature of pipe, F
t_2 = Initial temperature of pipe, F
L = Latent heat of steam at final temperature, Btu per lb
0.114 = Specific heat of wrought or steel pipe

Example: Find the amount of steam condensed when steam at 10.3 lb gage pressure is admitted to 500 ft of 5 in. welded piping at 70 F.

From any standard pipe table, the weight of 1 ft of 5 in. pipe is found to be 14.8 lb. From steam tables, $L = 952$ Btu. $t_1 = 240.1$ F. Then substituting in the equation:

$$C = \frac{(14.8 \times 500) \times (240.1 - 70) \times 0.114}{952} = 150.73$$

Table 1, showing the weight of steam condensed per foot of pipe, 2 in. to 15 in. in diameter, in raising pipe temperature from 70 F to steam temperature at common steam main pressures, has been prepared using the above formula. Since so much of the new steam piping is welded, the weight of pipe flanges has not been considered in the table.

Radiation Losses from Piping

After the pipe has reached steam temperature, additional steam will be condensed to make up for radiation losses from the pipe. At temperature differences up to 250 F between the bare pipe and the surrounding air, the heat loss is conservatively figured as being 3 Btu per hr per F temperature difference per sq ft of external pipe surface. The amount of condensate expressed as a formula is:

$$C = \frac{A \times 3 \times (t_1 - t_2)}{L}$$

where

C = Weight of steam condensed, lb per hr
A = External area of pipe, sq ft
t_1 = Steam temperature, F
t_2 = Air temperature, F
L = Latent heat of steam, Btu per lb

At temperature differences exceeding 250 F, the following heat loss figures may be used in the formula (instead of 3 Btu):

Temperature Difference, F	For 3 in. Pipe	For Large Dia. Pipe.
300	3.5	3.2
350	3.8	3.6
400	4.6	4.1
450	4.75	4.5
500	5.4	5.0

Example: What weight of steam will be condensed due to radiation losses from 500 ft of 5 in. bare pipe under 10 lb steam pressure with an air temperature of 70 F? From a pipe table, 0.686 ft of 5 in. pipe has 1 sq ft of external surface. Therefore, 500 ft of pipe would have an area of 500 ÷ 0.686 or 728.8 sq ft. Substituting in the formula,

*Armstrong Machine Works, Three Rivers, Mich.

$$\frac{728.8 \times 3 \times (240.1 - 70)}{952} = 391 \text{ lb per hr of condensed steam}$$

Table 2 is derived from the above formula and shows the amount of steam condensed to make up for radiation losses from one lineal foot of bare pipe, sizes 2 in. to 15 in., at common steam pressures.

The heat loss from, and hence condensation in, insulated piping is reduced from 60 to 85 per cent (depending upon thickness and efficiency) by insulation. For trap selection purposes, it is advisable to be conservative and figure that the steam condensed in insulated pipe to make up for radiation losses will be 40 per cent of the amounts shown in Table 2.

Spacing the Traps

Before calculating the warming up load and the normal load due to radiation losses from a given length of steam main, it is first necessary to determine where or how far apart the traps should be installed. Then, given the length and diameter of pipe to be drained by a single trap, it is easy to calculate the amount of condensate and size trap required.

Unfortunately, little is known regarding the flow of condensate in high velocity steam mains. One possibility is that when steam first condenses it forms a film of water all around the interior of the pipe. As the film thickness increases, most of the water flows to the bottom of the pipe forming a tiny stream of condensate. In spite of the high steam velocity, this small trickle of condensate at the bottom of the pipe moves rather slowly because of friction between the pipe and the water. However, as the depth of the stream increases the relative friction decreases and the trickle of condensate gains speed. With increased depth, too, waves will form on the surface of the condensate stream and still further increase its speed. Should these waves attain a size such as partially or wholly to block the passage of steam trouble will occur. With the steam flow retarded the steam pressure will drop ahead of the wave, resulting in increased wave velocity sufficient to damage the pipe.

Because of their high velocity, these waves or slugs will pass right by drip connections to steam traps. Large receiver separators can be installed of course, but the cost of these separators and the large traps required to drain them may be greater than the cost of smaller traps placed at sufficiently close intervals to prevent the formation of slugs.

Another theory is that as small ripples form on the condensate surface these ripples will be blown into a mist or spray by the high velocity of the steam, making the use of a separator absolutely necessary in order to insure dry steam at the point of delivery. Both theories may be correct, depending upon the velocity of the steam in the pipe, the steam pressure, and the size of the pipe. However, regardless of whichever theory may apply to a given installation, it seems obvious that traps should be installed at sufficiently close intervals to remove condensate before it can accumulate in slugs or be blown into a spray that complicates the condensate removal problem.

In important steam distribution projects constructed during the past few years, traps have been spaced at intervals ranging anywhere from 300 to 800 ft. A spacing of 500 ft therefore can be considered conservative, with a possible variation each way depending upon local conditions. In district heating work, for example, it is sometimes customary to install traps at every manhole station and the location of the manholes is governed by other considerations such as length of city blocks, pipe expansion, etc.

Regardless of how short the intervals are, traps should be installed at all natural drip points (such as ahead of inverted loops, at the bottom of regular loops, and at the bottom of supply risers). Traps also should be installed ahead of important reducing valves and ahead of shut-off valves, particularly when a water seal will form over the closed valve disc.

Picking the Right Trap Size

After determining the length of line to be drained by a single trap, either from the natural drip points or conservative maximum spacing, figure the amount of

Table 1—Steam Condensed per Foot of Pipe in Raising Pipe from Atmospheric to Steam Temperature

Steam Pressure, Lb per Sq In. Gage →		2.3	5.3	10.3	25.3	50.3	100.3	150.3	200.3	250.3
Steam Temperature, F →		219.4	228.0	240.1	267.3	298.0	338.1	366.0	388.0	406.2
Temperature Rise from 70 F →		149.4	158.0	170.1	197.3	228.0	268.1	296.0	318.0	336.2
Pipe Size, in.	Wt. Per Ft, Lb	Lb of Steam Condensed								
2	3.65	0.0643	0.0687	0.0745	0.0883	0.104	0.127	0.144	0.158	0.170
2½	5.8	0.102	0.109	0.118	0.133	0.165	0.202	0.232	0.250	0.270
3	7.6	0.133	0.143	0.155	0.183	0.217	0.264	0.299	0.328	0.354
3½	9.2	0.161	0.173	0.188	0.222	0.264	0.320	0.362	0.397	0.429
4	10.9	0.192	0.205	0.222	0.263	0.311	0.378	0.429	0.472	0.509
5	14.8	0.261	0.278	0.302	0.358	0.423	0.514	0.582	0.641	0.690
6	19.2	0.338	0.362	0.392	0.464	0.549	0.666	0.755	0.829	0.895
7	23.8	0.418	0.448	0.485	0.575	0.680	0.826	0.937	1.030	1.109
8	28.8	0.507	0.542	0.589	0.696	0.822	1.002	1.135	1.248	1.344
9	34.1	0.600	0.642	0.696	0.825	0.976	1.185	1.343	1.478	1.591
10	40.5	0.712	0.761	0.827	0.978	1.156	1.408	1.593	1.752	1.890
11	45.6	0.803	0.858	0.931	1.102	1.304	1.588	1.795	1.973	2.125
12	49.6	0.871	0.932	1.011	1.197	1.420	1.722	1.953	2.146	2.312
13	54.6	0.960	1.029	1.116	1.321	1.560	1.899	2.150	2.362	2.542
14	58.6	1.031	1.102	1.196	1.416	1.674	2.035	2.310	2.535	2.730
15	62.6	1.103	1.179	1.280	1.513	1.792	2.175	2.465	2.715	2.920

steam required to bring the pipe line up to steam temperature. This amount of condensate should be converted into an hourly flow basis by multiplying by 60 and dividing by the number of minutes required to bring the line up to temperature. For example, approximately 348 lb of steam are required to raise the temperature of 500 ft of 9 in. pipe from 70 F to the temperature of steam at 10.3 lb gage. If only six minutes time is required for bringing this pipe up to temperature, the hourly rate of flow would be $(348\times60)\div6$ or 3480 lbs.

To handle this warming up load flowing at the rate of 3480 lb per hr a trap must have a continuous discharge capacity somewhat in excess of this figure; in fact, a safety factor of at least 1½ or 2 would be advisable. That is a trap should be used the continuous discharge capacity of which is approximately 5,000 lb per hr at 10 lb gage pressure. The selection of a trap to handle 5000 lb of condensate per hr at 10 lb pressure would be very simple if all traps were rated on the basis of continuous discharge when handling condensate at steam temperature. Unless it is known that the ratings are so based, it is usually advisable to apply an additional safety factor, as a trap test made with cold water will show almost twice the capacity of the same trap tested with hot condensate at the same pressure differential.

In case steam is admitted to the pipe line more slowly, the warming up time will be greater and this of course reduces the maximum rate of flow. However, this does not necessarily mean that a smaller trap can be used because the longer warming up period means that the trap will have to operate for some little time at a pressure much below the maximum. The capacity of a trap at 1 lb pressure for example is approximately only half its capacity at 10 lb.

Now compare the warming up load with the amount of steam condensed to make up for the radiation losses from the pipe line. Referring to Table 2, 1 ft of 9 in. pipe will condense 1.354 lb of steam per hr at 10.3 lb gage steam pressure, or 677 lb per hr for the 500 ft line. This radiation loss load is much less than the warming up load and hence a trap large enough to take care of the initial flood of condensate has ample capacity to take care of the steam condensed to make up for radiation losses. Insulating the pipe line adds still further to the margin of safety for handling the steam condensed by the line after it is brought up to temperature.

Traps for draining steam supply risers are selected on the same basis as traps for horizontal steam mains. They should be trapped at the bottom. On a properly designed job, their size is such that condensate will flown down counter-current to the steam. Since the condensate film will be fairly uniform in depth all around the riser, a suitable drip pocket is essential at the bottom of the riser.

In the case of lines that are filled with steam at all times, it is possible to ignore the warming up load provided the initial load of condensate is taken care of by manually controlled valves at the end of the steam line. Here the steam condensed to make up for radiation losses governs the size of trap to be used and conservative practice requires that the continuous discharge capacity of the trap should be from two to three times the amount of steam that would be condensed even if the lines were not insulated. If the lines carry superheated steam, these same rules for trap selection should be used because there is always a possibility that the superheat may be lost and that condensation will be formed in sufficient quantity to cause damage unless it is trapped off promptly.

Getting Condensate to the Trap

Once the proper size of trap is selected for any given length of steam line, the next problem is to make sure that all the condensate from the header will enter the line to the trap. The common practice of merely screwing the drip line into the bottom of the steam header is hardly adequate as any sizable slug of condensate will be carried right past the drip connection by the high velocity steam. In the days when all steam pipe was flanged, the cost of special flanged fittings made the installation of drip pockets fairly expensive and to some extent probably accounts for the practice of merely connecting the trap line to the bottom of the steam header.

Table 2—Steam Condensed to Make up for Radiation Losses from One Foot of Pipe (not Insulated)

Steam Pressure, Lb per Sq In. Gage→		2.3	5.3	10.3	25.3	50.3	100.3
Steam Temperature, F→		219.4	228.0	240.1	267.3	298.0	338.1
Temperature Rise from 70 F→		149.4	158.0	170.1	197.3	228.0	268.1
Latent Heat, Btu per Lb→		956.6	960.0	952.0	933.3	911.0	879.8
Lb Condensed per Sq Ft of Surface→		0.465	0.494	0.536	0.635	0.750	0.914
Pipe Size, In.	Length per Sq Ft Surface	Lb per Hr of Steam Condensed					
2	1.608	0.290	0.307	0.334	0.395	0.467	0.568
2½	1.328	0.350	0.372	0.405	0.478	0.565	0.689
3	1.091	0.426	0.452	0.492	0.581	0.687	0.837
3½	0.954	0.466	0.518	0.563	0.673	0.787	0.958
4	0.848	0.524	0.583	0.633	0.749	0.885	1.078
5	0.686	0.678	0.720	0.781	0.925	1.092	1.332
6	0.576	0.808	0.858	0.932	1.101	1.301	1.587
7	0.500	0.930	0.988	1.072	1.270	1.500	1.828
8	0.442	1.050	1.118	1.212	1.437	1.700	2.066
9	0.396	1.174	1.249	1.354	1.602	1.895	2.305
10	0.355	1.310	1.391	1.510	1.790	2.153	2.575
11	0.325	1.430	1.520	1.650	1.953	2.307	2.810
12	0.299	1.555	1.652	1,794	2.121	2.510	3.055
13	0.27	1.720	1.830	1.996	2.350	2.780	3.381
14	0.25	1.860	1.975	2.145	2.540	3.000	3.655
15	0.24	1.940	2.060	2.235	2.645	3.125	3.805

Selection and Maintenance of Industrial Air Conditioning Systems

By **William A. Hanley***

IN any industrial application of air conditioning equipment, from the time the first line is drawn by the draftsman until the job is complete and running, the foremost thought of all parties concerned should be a system that will operate reliably and be easily and cheaply maintained. Most processes which justify air conditioning are important ones and cannot be interrupted without loss. Use of the best materials available and proper design are therefore important. Unreliable equipment is never satisfactory; no matter how great a saving there appears to have been made in the original investment.

After the problem has been analyzed by competent engineers and the job turned over to the designer, the purchaser will do well to check up on a number of important points.

The shell, eliminators and fan can be made of a number of materials. Black and galvanized iron are commonly used, but copper and stainless steel are available, and may be justified either from the point of contamination of the product to be air conditioned where iron or zinc are objectionable or from the matter of maintenance, where materials are indicated that will not be affected by acid or alkali reactions. In the final analysis there is one best metal to be used consistent with the job to be done, and that metal should be installed.

When the washer is to be externally insulated with cork or a similar material, the designer should plan in so far as possible for the removal of such insulation without its destruction, should repairs or replacements be necessary inside the washer. A year or two ago the writer replaced an entire washer shell wihout the loss of any of the 3 in. cork insulation.

Piping should be installed to be as accessible as possible. Ordinary galvanized steel pipe is commonly used, but copper bearing or wrought iron pipe—or even more expensive materials—are available and should be used when conditions justify.

Pumps usually are of cast iron, but where conditions require, can be of bronze and other slow oxidizing metals.

The control, recording and indicating instruments should be of the best. If quality is to be compromised somewhere, it should not be in the instruments. They are to the air conditioning system what the nervous system is to the human body and accuracy and reliability are of paramount importance.

A well designed and carefully installed plant should be conscientiously maintained. Recording instruments are essential and justified on any process which must have conditioned air. The daily or weekly charts should be changed regularly and inspected by a competent observer, and the recorders should be checked monthly for accuracy.

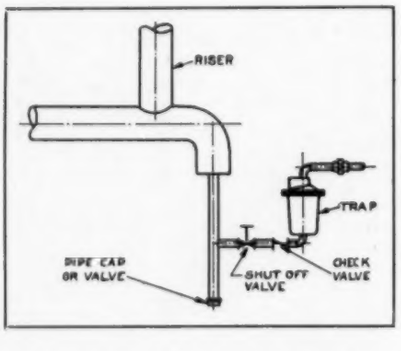
In most cases the washer screens should be changed daily, where stationary screens are provided, and the washer thoroughly flushed out at least once a week. The washer should be filled with clean water every two or three days.

The pump and motor lubrication should be checked at least weekly even though ring oil or ball bearings are in use. The checking is a matter of seconds only.

It may be necessary to clean spray heads every day, but usually for washing outside air, every week or two

[*Continued on page* 120, *back section*]

*Director of Engineering, Eli Lilly and Company, Indianapolis, Ind. Member of Board of Consulting and Contributing Editors.



Steam header drainage trap installation

Now that welded piping has become so common, it is a relatively simple matter to provide a drip pocket at the bottom of the steam header. This pocket should have some storage capacity to take care of any slugs that might come at a faster rate than the continuous discharge capacity of the trap, and furthermore such a pocket compensates for any lag in the opening of the trap itself. Where it is not feasible to install pockets in the line for drip connections, a small dam could be tack welded in the bottom of the pipe to serve as a baffle to divert condensate to the pipe connection. The height of the dam would not need to be more than one-tenth of the pipe diameter and the drip connection should be made on the upstream side of the dam. Wherever possible, a dirt pocket should be installed ahead of the trap to collect any scale or pipe cuttings that otherwise might enter the trap and interfere with its satisfactory operation.

Traps for draining steam headers should be non air-binding otherwise auxiliary air removal equipment or manual control must be provided. If the steam is turned on and off at rather frequent intervals so-called "blast traps" are advisable, especially at the end of the steam headers. These blast traps are either of the combination float and thermostatic trap type or inverted bucket trap type having an automatic air bypass incorporated in the bucket.

"What Air Conditioning Did for My Business"

By **Owners of Air Conditioning Systems**

MORE and more restaurants, stores, shops and other establishments are learning the value of air conditioning as a business builder. Actual experience of others being an important factor when the installation of an air conditioning system is under consideration, what it has accomplished in several specific instances is briefly reported here.

Supplementing good food and service, appearance and comfort are essential to success in the restaurant business, according to D. L. Toffenetti, head of the Triangle restaurant chain in Chicago. Air conditioning is thus a necessity if a restaurant is to compete successfully for patronage; by attracting more customers and making those customers feel like enjoying a real meal it quickly pays for itself.

Experience with air conditioning over a period of years has convinced the Triangle restaurant head that this is true. Some years ago, when the value of air conditioning was considerably less known than it is today, a system was installed in one of the chain. Now it is considered just as important a service as, for instance, the dish-washing equipment. Even for the restaurant at the World's Fair last summer, which was equipped to be used for only a few months, air conditioning was installed, and accordinging to Mr. Toffenetti, the investment paid for itself in increased business in the short time. "The luxury of yesterday becomes the necessity of today," to his way of looking at it and air conditioning falls in this category.

It is not enough to provide the equipment for keeping his patrons comfortable, he feels. The thought and money that go into the installation of the system should be brought to the attention of the customers so that they will realize just what has been done the better to serve them.

Mr. Toffenetti has some ideas about the modernization of restaurants which are of interest, particularly as air conditioning should be carefully considered in any project of this kind. According to him, the volume of business of a restaurant is at a high level for the first five to eight years. After that it drops off, not because the food or service is not as good, but just because patrons grow tired of the same old place, new customers are not attracted, newer eating places draw trade away. Every restaurant should therefore be modernized completely every five years, he believes, and the owner must allow in his prices a certain percentage to make possible replacement of his restaurant at the end of each such period. That many neglect to do this is cited by Mr. Toffenetti as the reason for the average life of a restaurant being five years. The Triangle organization is investing some $60,000 in modernizing one of its units at the present time.

Summer air conditioning is also a boon to the confectionery store, as proved by the experience of Mrs. Snyder, who operates several candy shops in Chicago and suburbs. She considers expenditures for air conditioning "one of the best investments" she has ever made in her business. The air conditioned Snyder shop (12 ft by 50 ft) at the World's Fair last summer attracted 2,800 customers the opening day, a record which was broken on a number of other days during the summer. In addition to attracting customers (some said admission should be charged just to come in and be cool), heavy losses in stock are avoided by summer air conditioning, doors and windows may be kept closed, dust and flies are kept outside and the shop is consequently more inviting.

It doesn't have to be a food product to need the stimulation of summer air conditioning. Gregory's (knitted sportswear for women), installed a unit last July in their Oak Park, Ill., store and "Impossible as it seems," says Mr. Gregory, "that month was better by 50 per cent than any we've had since we've been in business." The problem at this shop was to get women to try on knitted dresses in warm weather. As soon as the cooling unit was installed, customers began coming in, actually "stalled" to keep from going outside again. It is Mr. Gregory's belief the unit paid for itself in three weeks.

D. L. Toffenetti, head of the Triangle restaurants and famous for his descriptive menus, believes in air conditioning as a business builder. He also believes that the patron should be informed of the thought and investment that must be given so that he may be comfortable, as is indicated by the sign seen here. This view shows the glass-enclosed machine room and the 75 ton freon compressor at the Triangle restaurant which served thousands at the World's Fair last summer.

Suggestions for

MODERNIZING PLANT HEATING

MAINTENANCE of heating systems has been neglected during the past few years, as has the repair and replacement of other plant services. This has resulted in lowered efficiency and higher cost of heating.

What to do about it needs consideration at the present time. One reason is that with improved industrial activity, money is more easily available to invest in improved heating service and lowered operating costs. Too, the results secured by those plants which have modernized heating prove that the obsolete heating system offers a real opportunity for improvement. Heating is a major and essential service in plants as well as in residences; and the FHA loans have indicated that the heating plant is where modernization is most needed.

To aid those plant engineers and executives who have modernization of their heating services under consideration, the views of several heating experts are presented here. Each of these authorities is a member of H. P. & A. C.'s board of consulting and contributing editors.

Need for an Engineer's Survey

Homer R. Linn[1] believes that the managers of a great many industrial plants would do well at this time to employ consulting heating engineers to make thorough inspections of the heating plants and give the owners a complete report on what it is costing under existing conditions to heat the buildings.

This report should point out the places where the greatest losses are, due to the present system, and what changes should be made to overcome these losses. Then in conclusion, the report should point out what the cost would be in making changes in the system fully to overhaul it and what the profit would be in the operation of the system if it were so overhauled. Also, it would be of interest to the plant owner for the engineer to show what it would cost to make *partial improvements* and what savings would result from such changes.

There are so many new methods of heating today as compared with those of ten years or more ago, when many heating plants now in service were installed, that an engineer could correctly point out what these different methods would cost, what savings they would produce, and which one would be the best suited under existing conditions.

Proper Design Important

C. W. Kimball[2] stresses that the most important point in designing any new heating system is a proper and thorough study of the needs of the particular plant so that the final selection of the various parts entering into it may all fit together to produce the best results. Many times a lack of consideration of the facts lead to a badly proportioned whole. In bringing together the various parts of the plant, each item must be selected to insure the results desired. Engineers should be as careful to produce a reasonable first cost plant as they should be to

[1] Engineer, Western Executive Office, American Radiator Company, Chicago.

[2] Richard D. Kimball Co., Engineers, Boston.

produce an economically operating plant. He has come across at least two cases where the first cost of the plant designed by engineers was so excessive that the whole scheme was dropped and cheaper and questionable equipment was substituted.

In regard to possible heating wastes, there are improper combustion, waste of water, waste of heat due to leaks, poor covering, faulty piping, leaky traps, too small piping causing excessive repairs, dirty boilers, poorly designed and leaky settings, poor draft, too much draft, too much and unnecessary pressure on the heating systems, lack of attention to small wastes and finally lack of adequate records to tell what is being done.

The list of new equipment suitable for improving plant heating is so long and varied that an engineer may be led astray in his selection. It seems that better understanding of combustion, better instruments, better fuel burning devices, better heaters, coverings, better operation, and a better cooperation between the plant engineers and the "office" all have a decided effect on better results.

Adaptability of Unit Heaters to Plants

In the design of new heating systems for industrial plants there are three major considerations, all of which are met by suspended type unit heater systems, according to A. W. Moulder[3]:

Adaptability: The ability of the system to fit into the physical conditions to best advantage.

[3] Manager, Heating-Power and Industrial Piping Division, Grinnell Company, Inc., Providence, R. I.

The writer's experience indicates that with a system of unit heating the piping may be installed to fit plant conditions and processes. Because the units are overhead, both the supply and return piping may also be well up, out of the way, and the burying of return piping eliminated.

Efficiency of Operation: This takes into account uniformity of heating to meet conditions, as well as economy of operation.

The fact that unit heaters blow the heated air where it is useful and at the same time, if properly designed and installed, maintain uniform temperatures as required, make for the most efficient type of heating.

The control of the suspended type unit heaters is of very great importance if real economy is to be had. The fans on the units may be controlled or the steam to the units may be controlled. In some cases, it is advantageous to control both fan and steam. It is also desirable to consider the question of day and night time control, *i. e.*, maintaining a lower night-time than daytime temperature. In the event steam is supplied directly by boiler the proper consideration for control of this boiler is of outstanding importance.

Cost: This represents both first cost and depreciation and maintenance expense.

The first cost of such a system will be low.

The simplicity of the unit heating system makes it an easy and economical system to maintain because of less piping, fewer joints, fewer valves, traps and specialties. The only possible item in the unit heating system which

[*Continued on page* 120, *back section*]

LARGEST CONSTRUCTION PROJECT in Chicago since 1929 —except for the World's Fair—was the rebuilding of the Union Stock Yards after the fire of May 19, 1934; the rebuilding was completed in time for the International Live Stock Exposition, held last month. This view looks east from the top of the 9 story Exchange building

The rebuilding involved industrial, commercial and office, hotel, restaurant, theater and exhibit buildings. The industrial type buildings are heated by unit heaters with the exception of small spaces where direct radiators were required.

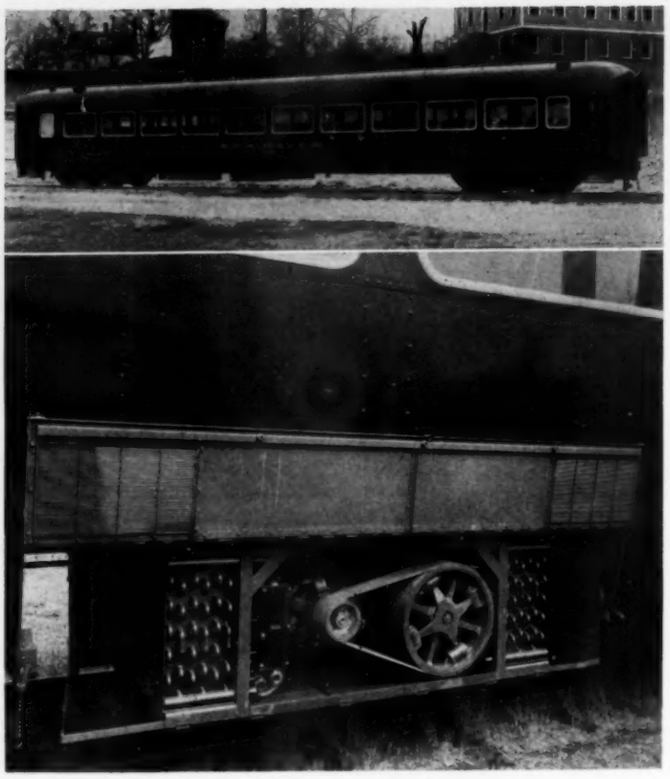
The commercial and office buildings and the hotel building are heated by direct radiators; ventilation systems are installed where needed. The new restaurant building is of "air-tight" construction and is completely air-conditioned.

The International Amphitheatre (seen at the right) is heated and ventilated by four separate blast systems, with the exception of the exhibit halls where unit heaters were employed.

Vacuum heating systems are used in all the buildings, steam being furnished from a central power house. The main vacuum pump is in the power house, but there are auxiliary pumps at different points on the premises. In some cases the source of steam supply is practically one mile from the building heated. A. Epstein, structural engineer, designed and supervised all buildings with Walter A. Becker of his staff in charge of mechanical installation.

Railway Air Conditioning Makes Rapid Strides

New Haven Sets New Standards



The new air conditioned, modern New Haven coaches with close-up of air conditioning equipment

REMARKABLE advances were made in the application of air conditioning to railway passenger equipment during the past year, the railroads finding comfort for their patrons to be a business stimulator. Plans for 1935 include installation of air conditioning systems in hundreds of additional cars; entire trains will be air conditioned where previously only diners, club cars, etc., have been so equipped.

The New Haven Railroad (for example) has set a new standard in passenger equipment with the air conditioned cars it is now putting into regular day-coach service on its through runs. Of a modified streamline variety (styled by Walter Dorwin Teague) the new cars, 12 of which were in service before Christmas, the remaining 32 coming along from day to day, are a radical departure from ordinary coaches. They are built of high-tension alloy steel which cuts down weight by about 20 per cent, tend to the tubular in shape.

Interiors are striking, with walls of french grey, light blue upholstery, seats framed in stainless steel, aluminum baggage racks, and an off-white ceiling with three vermilion stripes running the length of the car to lend a gay touch. Soft, deep-cushioned seats are set low with the backs at a slightly reclining angle, provide more width and leg room. They are equipped with an unusual swivel arrangement whereby they may be swung around to provide facing seats for bridge or sociability, and automatically lock into place.

Windows have double plate glass with a dehydrated pocket in between to act as insulation and prevent frosting or steaming up of the windows. Window sills are of a highly polished black finish which will not burn or blister. The windows are sealed in aluminum frames.

Quietness is one of the outstanding characteristics of the cars, is achieved by liberal use of rubber packing throughout the car, and by the sealing of the windows. Floors are cork board covered with an attractive linoleum of checkered design in two tones of dark blue.

The new type of lighting furnishes direct illumination for each seat at the reading angle, but eliminates the eye-strain of looking down the car and seeing a line of lights; the lights are recessed into the ceiling, the rays being directed through a square grille.

The air conditioning system has a low velocity air diffusing system which insures positive but gentle diffusion of conditioned air throughout the car, accomplished without marring the interior beauty of the cars in the slightest degree. The air conditioning distribution duct runs the length of the car along the ceiling and adds to the effectiveness of the lines which appear to give the car additional length.

The cars seat 84 passengers (four more than the last type of coach ordered by the New Haven), are used in regular trains between New York and Boston, New York and Springfield, Hartford, and Waterbury, with no extra fare.

The now-famous Chesapeake & Ohio's kitten dramatizes the comfort of this road's air conditioned cars

Heating·Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers JOURNAL SECTION

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Study of Summer Cooling in the Research Residence for the Summer of 1934

By A. P. Kratz*, S. Konzo**, M. K. Fahnestock***, and E. L. Broderick† *(MEMBERS)*
Urbana, Ill.

This paper is the result of research conducted at the University of Illinois in cooperation with the A. S. H. V. E. Research Laboratory and the National Warm Air Heating and Air Conditioning Association

The results presented in this paper were obtained in connection with the summer cooling investigation (1934) in the Research Residence,[1] Fig. 1, at the University of Illinois, conducted by the Engineering Experiment Station of which M. L. Enger, Dean of the College of Engineering, is the director, in the Department of Mechanical Engineering of which O. A. Leutwiler, Professor of Mechanical Engineering Design, is the head. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Special acknowledgment is due to the General Electric Co., the Utilities Research Commission of Chicago, Ill., and the Frigidaire Corp. for active cooperation in this investigation. Acknowledgment is also due to A. F. Hubbard, Graduate Assistant, for active participation in the detailed work of the investigation.

PREVIOUS investigations in summer cooling in the Research Residence during the summers of 1932[2] and 1933[3] made use of ice as the medium for cooling water circulated through coils placed in the forced-air heating system, and were confined largely to studies of the factors affecting the cooling load, and to studies of the effectiveness of circulating air from the outdoors at night, both as a supplement to artificial cooling during the day and as a means for eliminating the necessity for artificial cooling during the day. In the course of these studies no outdoor air for ventilation was taken into the Residence during the periods when artificial cooling was required. The investigation for the summer of 1934 was undertaken to determine to what extent a two-ton mechanical refrigerating unit could be used to produce satisfactory cooling in the Research Residence, both when supplemented by the circulation of outdoor air through the second story at night, and when not so supplemented, under conditions in which approximately one air change per hour of outdoor air was used for the purpose of ventilation during the periods when artificial cooling was required.

Fig. 1—View of Research Residence in Urbana, Illinois

Description of the Research Residence and Cooling Equipment

The Research Residence, shown in Fig. 1, together with the forced-air heating system has been described in a previous paper.[2] For the purpose of this investigation, the Residence was equipped with awnings at all east, south, and west windows, and the sun parlor was isolated from the rest of the house by means of the doors opening into the dining room. The entire third story was regarded as an attic, and during the daytime was isolated from the rest of the house by means of a door at the head of the stairs. The attic windows, however, were opened to provide ventilation in the attic during both day and night. With the exception of the space above the northwest bedroom and the small spaces adjacent to the dormer windows, the third story had hardwood floors laid on pine sub-flooring. The small spaces adjacent to the dormer windows had no floors. In the space above the northwest bedroom 1 in. of insulating blanket was nailed to the upper edges of the floor joists. Hence, practically all second floor ceilings

*Research Professor, Engineering Experiment Station, University of Illinois.
**Special Research Associate, Engineering Experiment Station, University of Illinois.
***Research Assistant Professor, Engineering Experiment Station, University of Illinois.
†Research Assistant, Engineering Experiment Station, University of Illinois.

[1] The Research Residence in Urbana, Ill., was built, furnished, and completely equipped specifically for research work in warm-air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

[2] A. S. H. V. E. Research Paper entitled, "Study of Summer Cooling in the Research Residence at the University of Illinois," by A. P. Kratz and S. Konzo. (*Heating, Piping and Air Conditioning*, February, 1933, pp. 115-126.)

[3] A. S. H. V. E. Research Paper entitled, "Study of Summer Cooling in the Research Residence for the Summer of 1933," by A. P. Kratz and S. Konzo. (*Heating, Piping and Air Conditioning*, December, 1933, pp. 613-625.)

For presentation at the 41st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buffalo, N. Y., January, 1935.

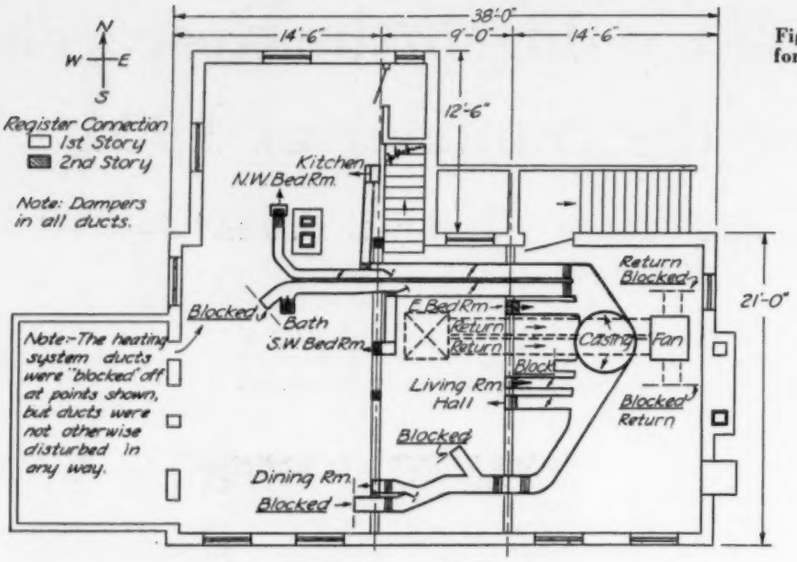


Fig. 2—Basement plan showing duct layout for forced-air system in use for distributing cool air

were at least equivalent to lath and plaster with flooring above it. No cooking was done in the kitchen, but the heat transmitted through the glass doors from the sun parlor, which was not ventilated by opening the windows, compensated for this to a certain extent. Unless otherwise specified, the state of the Residence was comparable for the work done during the three summers.

The arrangement of the forced-air heating plant and fan is shown in Fig. 2. For the purpose of this investigation, all return ducts with the exception of the central one containing the cooling coil were blocked. The delivery ducts to the sun parlor and third story were blocked, as indicated, and the dampers in the ducts to the first and second stories were adjusted to maintain the proper balance between the cooling on these two stories.

The arrangement of the cooling plant is shown in Fig. 3. The condensing unit consisted of a double pipe condenser and a four-cylinder compressor driven by a 3 hp motor. This unit was self-contained. Cooling was accomplished by direct expansion of Freon in an evaporator unit placed in a by-pass in the central cold air return duct. The evaporator unit or cooling coil consisted of 32 rows of finned copper tubing placed 4 rows deep in the direction of air flow and 8 rows high measured along the vertical axis of the air duct. The tubes were ¾ in. outside diameter and had 6 fins per inch. The overall diameter including fins was 1½

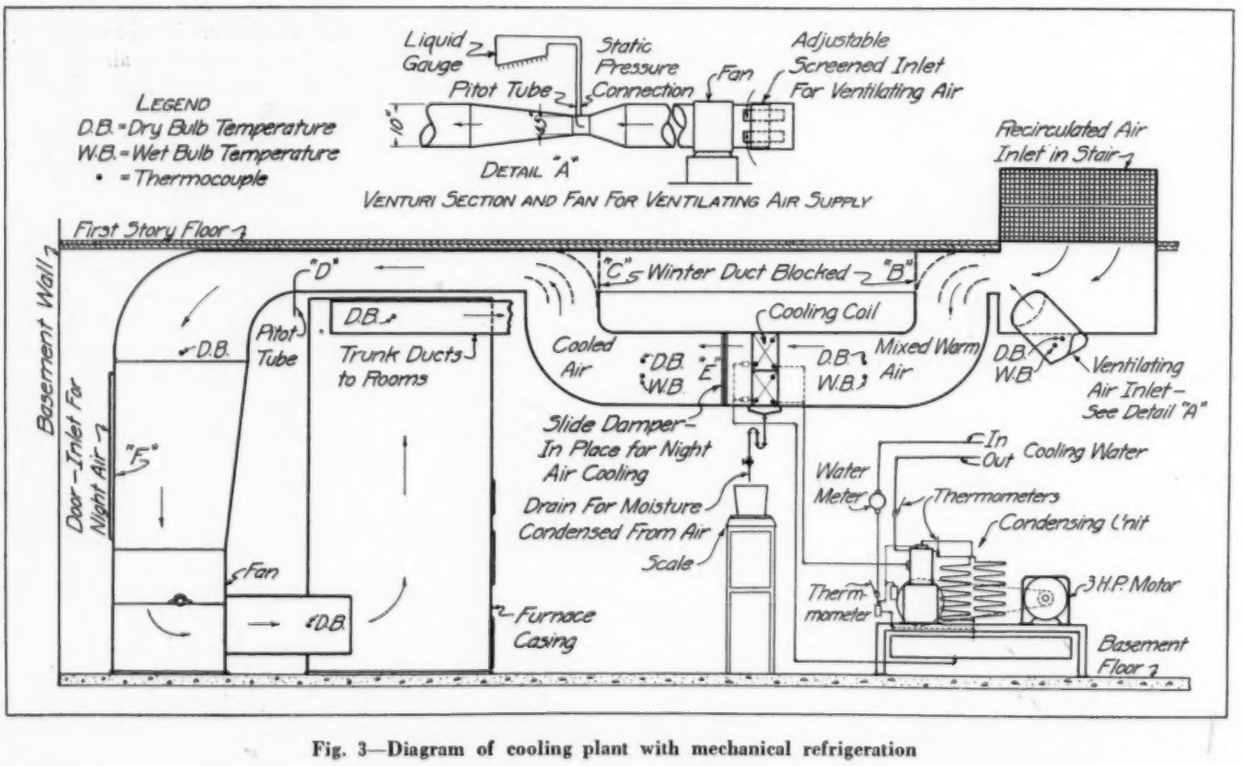


Fig. 3—Diagram of cooling plant with mechanical refrigeration

in. The evaporator as installed in the duct was 17 in. high and 30 in. wide, and the gross or face area presented to the flow of air was 3.54 sq ft, while the net or free area was 2.18 sq ft. The nominal rating of the condenser and evaporator units as used was 21,700 Btu per hour with refrigerant temperature of 32 F, water temperature of 80 F, and ambient air temperature of 100 F, or 29,500 Btu per hour with refrigerant temperature of 32 F, and water temperature of 60 F, with an air velocity of approximately 350 fpm across the 3.54 sq ft of face area.

The operation of the refrigerating machine was controlled by means of thermostatic expansion valves having the thermostatic bulbs clamped to the tubes of the evaporator; and a water control valve so designed that the degree of opening for the flow of the condenser water was controlled by the condenser pressure. The latter also operated to shut off the supply of water to the condenser when the machine was not running. The operation of the cooling plant as a whole was controlled by means of a room thermostat placed in the hall on the second story, and which served to start and stop the refrigerating machine in accordance with the cooling load required to maintain constant room temperature.

In order to provide for both heating in the winter and cooling in the summer, the evaporator, or cooling unit, was installed in a by-pass in the central return duct, as shown in Fig. 3. For the summer work, the duct was blocked with tightly fitting dampers at *B* and *C*, and all of the air delivered to the fan in the forced-air system passed through the cooling coil when the air in the house was being recirculated. For the purpose of providing outdoor air for cooling during the night, a slide damper, *E*, was placed in the by-pass on the downstream side of the cooling coil, and a door, *F*, was placed in the recirculating duct just ahead of the fan. When outdoor air was required, the basement door and the door in the recirculating duct were opened, and the slide damper was closed. The fan in the forced-air system delivered approximately 1,300 cu ft of air per minute when recirculating the air in the house and 2,200 cfm when using outdoor air at night. Outdoor air, for the purpose of ventilation during the periods when the cooling plant was operating, was provided by means of the duct shown as Detail *A* in Fig. 3. This duct contained a venturi section for the measurement of the volume of air delivered, and it was necessary to use a small fan in order to deliver the equivalent of approximately one air change per hour. Wet- and dry-bulb temperatures of the air and inlet and outlet temperatures of the condenser water were measured by means of thermocouples or thermometers placed at points indicated in Fig. 3.

Method of Conducting Tests

During the summer from June 5 to September 16, continuous records were made, by means of temperature recorders, of the following air temperatures: outdoor, dining room, kitchen, first story hall, living room, east bedroom, southwest bedroom, northwest bedroom, and third story hall. Continuous records were also made of the wet- and dry-bulb temperatures of the air entering and leaving the cooling unit, and of the outdoor air taken in for ventilation. Other incidental air temperatures were observed at regular intervals. Relative humidities both indoors and outdoors were observed by means of an aspirating psychrometer. The outdoor dry-bulb reading on this psychrometer served as a basis for correcting the outdoor temperature read from the recorder chart. Surface temperatures of the north, south, and west walls and of the northwest bedroom, east bedroom, and living room ceilings were obtained by means of thermocouples. The data were plotted on a continuous chart, and this chart was used as a basis for the analysis of the results.

During the periods of operation, observations were made of the weight of water circulated through the condenser and water jackets on the compressor, the temperature of the water entering and leaving the condenser and jackets, the temperature of the refrigerant entering and leaving and at the thermostatic control bulb in the evaporator, the head and suction pressures, and the electrical inputs to the compressor and fan motors. The weight of condenser water was obtained by means of a calibrated water meter and the air quantities were obtained from traverses made with Pitot tubes at section *D* in the central return duct and at the venturi section in the ventilating air duct as shown in Fig. 3.

During all of the tests, both with and without night cooling, the windows on the first story remained closed. The windows in the attic, with the exception of one opposite the door at the top of the stairs, remained open. For the purpose of night cooling, 11 windows on the second story were opened by raising the lower sash to the full extent. Two windows which were opposite registers and one window at the second floor stair landing remained closed. For all tests the Residence was operated strictly on the schedules described in connection with the following enumeration of the different test series:

Series 3-34. Artificial Cooling During the Day Supplemented by Circulation of Outdoor Air at Night

The second story windows and the attic and basement doors were closed at 7 a.m. and the fan, which had been delivering outdoor air through the system, was stopped. When the temperature of the indoor air on the second story rose to 81 F, the cooling plant was started, with the fan delivering through the cooling unit both the recirculated air and the outdoor air admitted for ventilation. The former was equivalent to 4.5 recirculations of the air in the house, and the latter was equivalent to one air change per hour, making a total of 5.5 air changes per hour delivered by the fan. The fan was run continuously through both *on* and *off* periods of the refrigerating unit. The cooling plant was allowed to operate with thermostatic *on* and *off* control maintaining 81 F on the second story until the effective temperature outdoors became equal to the effective temperature on the second story indoors. The refrigerating unit was then stopped; the second story windows and attic door were opened; and the dampers were set and the basement door opened, so that the fan delivered outdoor air through the duct system, the fan continuing to run until 7 a.m. The fan delivery was 2,251 cfm or 9.5 air changes per hour.

Series 4-34. Artificial Cooling During the Day Not Supplemented by Circulation of Outdoor Air at Night

The second story windows were not opened during the 24 hours. The fan was run continuously during the 24 hours, delivering through the cooling unit both the recirculated air and the one air change per hour admitted for ventilation. The cooling plant was allowed to operate with thermostatic *on* and *off* control maintaining 81 F on the second story.

Series 5-34. Circulation of Outdoor Air at Night Without Artificial Cooling During the Day

The second story windows and the attic and basement doors were closed at 6 a.m. and the fan was stopped. The house remained closed during the day, with no attempt made to control the indoor temperature. The second story windows and the attic and basement doors were opened at 6 p.m., and the fan was started. During the night, the fan took air from outdoors and delivered it into the rooms through the duct system. The fan delivery was 2,251 cfm, or 9.5 air changes per hour.

Series 6-34. Circulation of Outdoor Air at Night Without Artificial Cooling During the Day

Same as *Series 5-34* except that the windows were closed and the fan stopped at 7 a.m. instead of 6 a.m.

Results of Tests

Cooling with Outdoor Air at Night

During the summer of 1933, studies were made[3] with various combinations of window openings and fans for cooling with outdoor air at night. With one exception these studies were made with both first and second story windows open. The one exception was made with a fan in the attic drawing in 3,980 cfm which was approximately 33.6 air changes per hour based on the second story alone, or 16.8 based on the house as a whole.

Since it is undesirable in the average city residence to have the first story windows open all night, it seemed more practical in the case of the 1934 studies to confine the open windows to the second story alone. Furthermore, it also seemed more practical from the standpoint of the householder to employ the fan in the forced-air heating system circulating 2,251 cfm or approximately 9.5 air changes per hour, rather than to install additional fans in order to circulate a greater volume of air. The effectiveness of night air cooling obtained under these conditions, as compared with the effectiveness of cooling for the basement fan with both first and second story windows open, and for the attic fan with the second story windows only open, is shown in Fig. 4.

In the previous paper[3] it was shown that the minimum indoor temperature attained at night was practically the same as that existing when the windows were closed at 6 a.m. Furthermore, it was shown that the effectiveness of different methods of night cooling could be compared by plotting the drop in indoor temperature from the time that the indoor and outdoor temperatures became identical to the time at which the indoor temperature attained a minimum, against the drop in outdoor temperature from the time at which the indoor and outdoor temperatures became identical to the time at which the outdoor temperature attained a minimum. These temperature drops are shown as *B* and *A* respectively in the insert in Fig. 4. In comparing different methods of night cooling, temperature drop curves giving greater slopes indicate greater effectiveness of cooling, and the maximum possible effectiveness would be represented by a 45 degree line, or one having a slope of 1.0 when the two temperature drops are plotted to the same scale.

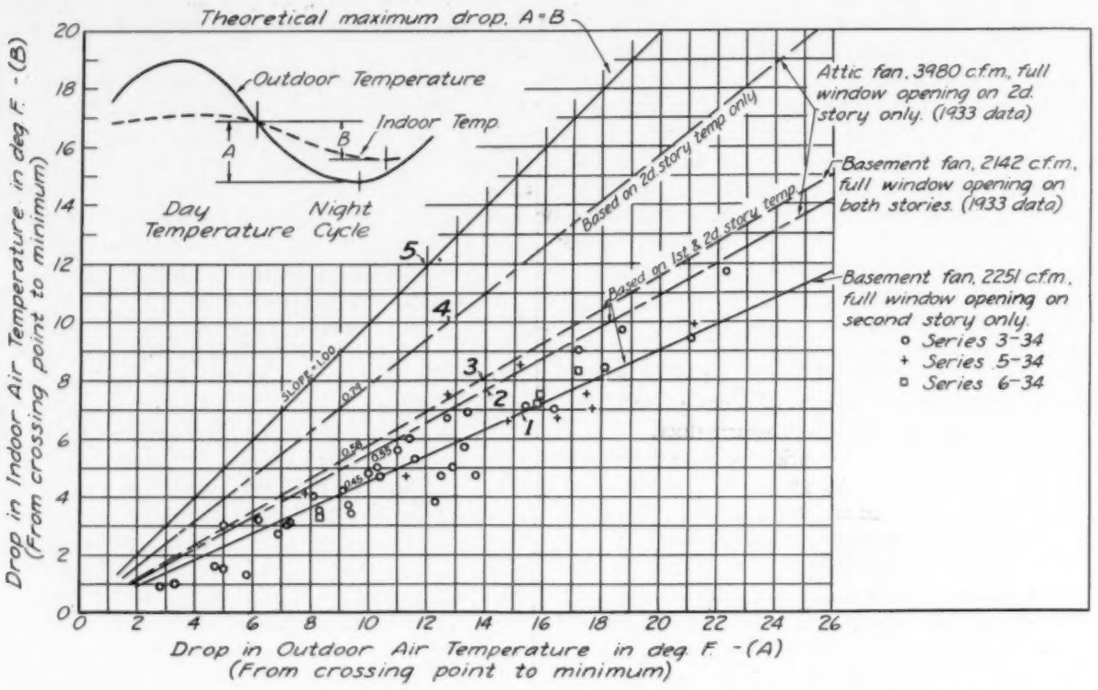

Theoretical maximum drop, A=B
Outdoor Temperature
Indoor Temp.
A
B
Day
Night
Temperature Cycle
Attic fan, 3980 c.f.m., full window opening on 2d. story only. (1933 data)
Basement fan, 2142 c.f.m., full window opening on both stories. (1933 data)
Basement fan, 2251 c.f.m., full window opening on second story only.
o Series 3-34
+ Series 5-34
□ Series 6-34
Based on 2d. story temp only
Based on 1st & 2d. story temp
SLOPE = 1.00
0.75
0.50
0.25
Drop in Indoor Air Temperature in deg. F. - (B) (From crossing point to minimum)
Drop in Outdoor Air Temperature in deg. F. - (A) (From crossing point to minimum)
0 2 4 6 8 10 12 14 16 18 20 22 24 26

Fig. 4—Temperature drops for various methods of night air cooling in Research Residence

From Fig. 4 it may be observed that in the case of the attic fan drawing approximately 33.6 air changes through the second story windows alone, the effectiveness in cooling the house as a whole, based on the average indoor temperature for both stories (Curve No. 2), was much less than that in cooling the second story alone based on the average indoor temperature for the second story (Curve No. 4). When cooling with outdoor air at night is used to supplement artificial cooling during the day, however, the effectiveness of night cooling in the house as a whole is of more significance than the effectiveness for the second story alone. The effectiveness of cooling the house as a whole was but slightly

greater for the fan in the forced-air heating system circulating approximately 9.5 air changes per hour and with both first and second story windows open (Curve No. 3) than it was for the attic fan drawing in 33.6 air changes per hour through the second story windows alone (Curve No. 2). The effectiveness of cooling for the house as a whole with the fan in the forced-air heating system circulating 9.5 air changes per hour and the second story windows alone open (Curve No. 1) was somewhat less than that for either of the latter methods. The location of the points representing the test data indicates that the effectiveness of night cooling was not greatly influenced by whether the windows were closed at 6 a.m., as for *Series 5-34*, or at 7 a.m., as for *Series 6-34;* or whether they were opened at 6 p.m. or at the time that the effective temperature became the same indoors and outdoors, as for *Series 3-34*.

It has been mentioned that the slopes of the curves similar to the ones shown in Fig. 4 afford a measure of the relative effectiveness of the different methods employed for cooling with outdoor air at night. Furthermore, the relative effectiveness of the different methods may be considered as dependent on the amount of air circulated and not inherent in the particular method used for circulating the air. Under these conditions, sufficient data are available from the studies of 1933[a] and 1934 to establish the range of the number of air changes that would be required to prove effective for night cooling in a house of the type of the Research Residence.

The slopes of all of the curves obtained when the outdoor air was positively circulated by means of some type of fan are shown in Fig. 5, plotted against the corresponding amounts of air circulated expressed as a number of air changes per hour. Two curves were thus obtained. The upper curve represents conditions either in a two-story house in which windows are opened on both stories and the effectiveness of cooling is based on

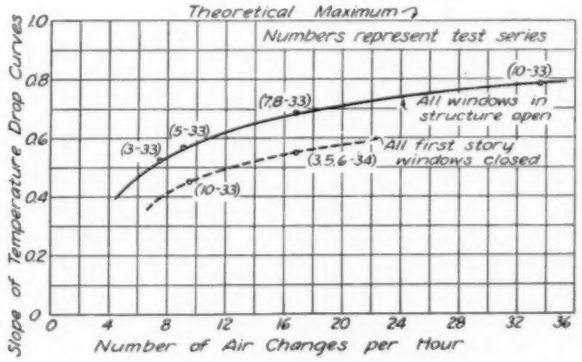


Fig. 5—Influence of rate of circulation of night air on cooling effect in the Research Residence

the average indoor temperature for both stories; or in the second story of a two-story house in which only the windows on the second story are opened and the effective cooling is based on the average indoor temperature of the second story alone. The fact that all of the points representing the data fell on a smooth curve, even though the amounts of window opening were not the same, indicates that the effectiveness of cooling for a given number of air changes per hour was more or less independent of the number and location of the windows opened, provided that the distribution of the openings was such as to give good distribution of the air. Furthermore, the fact that the point representing the cooling effect with 33.6 air changes per hour, obtained from consideration of the second story only, is on the curve, indicates that this type of curve is applicable to any space when the number of air changes per hour, the amount of window opening, and the indoor air temperature are considered in relation to the given space alone.

The lower curve in Fig. 5 represents conditions in a two-story house in which the windows on the second floor only are opened and the effectiveness of cooling is based on the house as a whole or on the average indoor temperature for both stories. It is evident that the effectiveness of cooling the house as a whole is less when the windows on the first story are not opened than it is when the windows on both stories are opened.

From Fig. 5 it may be observed that with the smaller numbers of air changes per hour the curvature of the curve decreases very rapidly as the number of air changes increases, but that the rate of decrease in the curvature becomes much less for the larger numbers of air changes. A slope of 1.0 would represent the maximum effectiveness of cooling, inasmuch as it would represent conditions under which the indoor temperature would be reduced to the same value as that outdoors. Within the limits of the observed data, the slope curve does not become asymptotic to a line representing a slope of 1.0, indicating that it would require air quantities far in excess of 33.6 air changes per hour in order to reduce the indoor temperature to the same value as that outdoors. It is therefore evident that cooling with outdoor air at night does not become reasonably effective until the number of air changes per hour reaches a value of approximately 9, and that above a value of 30 air changes per hour the gain resulting from increasing the amount of air circulated is very small. At some point the gain obtained by increasing the number of air changes per hour will probably be offset by the increased cost of electrical current required to operate the fan.

Cooling with Mechanical Refrigeration

The operating characteristics of the cooling plant and a comparison of the actual and calculated cooling loads for the house can best be illustrated by the results obtained on a typical day. For this purpose a test made on June 27, 1934, was selected and the results are shown in Fig. 6 and Table 1. On this day the outdoor temperature was 97.1 F at 2 p.m. and reached a maximum of 99.0 F at 3:45 p.m. At that time the house was being operated on *Series 4-34* and the windows had remained closed during the night preceding the test. The fan in the forced-air system had continued to operate during the *off* period of the compressor, shown from 1:30 a.m. to 6:30 a.m. in Fig. 6, and ventilating air from the outdoors was drawn in, resulting in a rise in the relative humidity indoors. At 6:30 a.m. the compressor was started through the action of the thermostat and operated intermittently until 11:20 a.m. The compressor operated continuously at full load capacity from 11:20 a.m. to 11:15 p.m., at which time intermittent operation was resumed. During the long *off* period from 1:30 a.m. to 6:30 a.m., the cooling coil evidently warmed up and the temperature became practically the

Table 1—Typical Operating Data and Results with Mechanical Refrigeration at 2:00 P. M. on June 27, 1934[a]

1.	Outdoor air	D.B. 97.1 F, W.B. 73.6 F, R.H. 33%
	Moisture content, gr per lb dry air	86.8
2.	Indoor air, avg. breath. level temp	1st Story 79.4 F, 2nd story 80.4 F
3.	Indoor air, house avg. at breath. level	D.B. 79.9 F, W.B. 65.2 F, R.H. 45.5%
	Moisture content, gr per lb dry air	70.0
4.	Ventilating air	D.B. 94.3 F, W.B. 76.5 F, R.H. 44.5%
	Moisture content, gr per lb dry air	108.6
5.	Mixed air entering cooling coil	D.B. 81.5 F, W.B. 67.2 F, R.H. 47.5%
	Moisture content, gr per lb dry air	76.9
6.	Mixed air leaving cooling coil	D.B. 65.6 F, W.B. 59.9 F, R.H. 72.0%
	Moisture content, gr per lb dry air	68.1
7.	Air temp. drop through cooling coil, F	15.9
8.	Temp. of cooled air leaving registers, 1st and 2nd story avg, F	70.0
9.	Air temp. rise in ducts and casing, F	4.4
10.	Basement air temp. at breath. level, F	77.3
11.	Quantity of air circulated	1,303 cfm or 5,660 lb of dry air per hour
	Density of air lb per cu ft	0.0724
12.	No. of house air recirculations per hour	5.5
13.	Cooling coil	face area 3.54 sq ft, free area 2.18 sq ft
	Air face velocity, fpm	368
	Air velocity through free area, fpm	598
14.	Ventilating air	240 cfm or 1,015 lb dry air per hour
	Density of air lb per cu ft	0.0707
15.	Moisture condensed from air, lb per hour	7.11
16.	Heat absorbed by cooling coil; total	29,065 Btu per hour
	Heat due to moisture in air	7,466 Btu per hour; 25.7% of total heat absorbed
	Sensible heat	21,599 Btu per hour; 74.3% of total heat absorbed
17.	Water temp. through condenser, F	Inlet 59.7, Outlet 98.0, Rise 38.3
18.	Water temp. through condenser, compressor and basement, F	Entering 59.7, Drain 100.0, Rise 40.3
19.	Quantity of condenser water, per hour	800.5 lb or 96.1 gal
20.	Heat absorbed by water passing through condenser	30,659 Btu per hour
21.	Heat absorbed by water passing through condenser, compressor and basement	32,262 Btu per hour
22.	Heat equivalent of power input to compressor motor	9,980 Btu per hour
23.	Net cooling load or heat gain on entire house (Item 21—Item 22)	22,282 Btu per hour
24.	Ratio of net heat gain to total heat absorbed in cooling coil (Item 23/Item 16)	0.767
25.	Temp. of cooling coil, F	Inlet 42.4, Control bulb 57.1, Outlet 74.0
26.	Refrigerant pressures, lb per sq in	Suction 40, Discharge 130
27.	Nominal rating of condensing unit with 32 F refrigerant, 60 F condensing water and 90 F ambient air	29,500 Btu per hour
28.	Compressor motor data	Size 3 hp, Measured power rate 2.92 kw
29.	Compressor speed	235 rpm
30.	Recirculating fan data	Speed 568 rpm
	Motor	Size 1/3 hp, Measured power rate 0.340 kw

[a]Test No. 7-34.

same as the air temperature. The intermittent periods of operation were not sufficiently long to establish equilibrium, and full capacity of the machine was not absorbed from the air until the compressor started to operate continuously.

During the hours from midnight to 5 a.m. the indoor temperature remained constant owing to the heat capacity of the structure while the calculated load curve showed decreasing cooling load. From 6 a.m. until 10 a.m. the actual average load approximated the calculated load. After 10 a.m., however, the actual load was less than the calculated load until approximately 6:10 p.m., at which time the two became equal. The calculated load attained a maximum of 40,300 Btu per hour while the actual load never rose above 30,500 Btu per hour. The effect of heat lag of the structure is most strikingly shown in the period from 6:10 p. m. to 11:15 p.m. At the latter time the calculated load had decreased to 12,500 Btu per hour while 27,500 Btu per hour were still being absorbed from the air. During the day the temperature of the air indoors rose from 79 F to 81 F. From 6:10 p.m. to 11:15 p.m. the indoor temperature was reduced from 81 F to 79 F. This was taken into account in the calculated load curve but is not sufficient to explain the wide discrepancy between the actual and calculated loads. From 11:15 p.m. to 3 a.m. the calculated load continued to decrease somewhat more rapidly than the actual load.

The calculated cooling load was based on the first and second stories only. It included the heat transmission through the walls, floors, ceilings, and glass, sun effect on walls and glass, sensible heat brought in by the air used for ventilation, heat brought in by moisture in the air used for ventilation, heat from 4 occupants, heat from lights, and heat supplied by the electrical input to the fan, and was based on the hourly observed indoor and outdoor wet- and dry-bulb temperatures. The heat equivalent to the electrical input to the compressor motor was not included as it was regarded as all being dissipated in the basement. An attempt was made to determine the pressure built up in the first and second stories by the operation of the ventilating fan. While this did not result in a quantitative determination of the actual pressure, it did indicate that the pressure was slightly greater than atmospheric. Hence it was considered that infiltration through windows could be neglected and all infiltration could be credited to the air brought in for ventilation, amounting to approximately one air change per hour.

The sun effect based on unshaded windows was calculated by using the sun intensity curves for east, south and west vertical surfaces for the month of July from the work of F. C. Houghten[4] and others, and by multiplying the value thus obtained by 0.83[2] to obtain the heat transmitted through the glass. Since all of the windows in the walls exposed to the sun were equipped with awnings, the heat transmitted through the shaded win-

[4]A. S H. V. E. Research Paper entitled, "Heat Transmission as Influenced by Heat Capacity and Solar Radiation," by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott. (TRANSACTIONS A. S. H. V. E., Volume 38, 1932, pp. 231-279.)

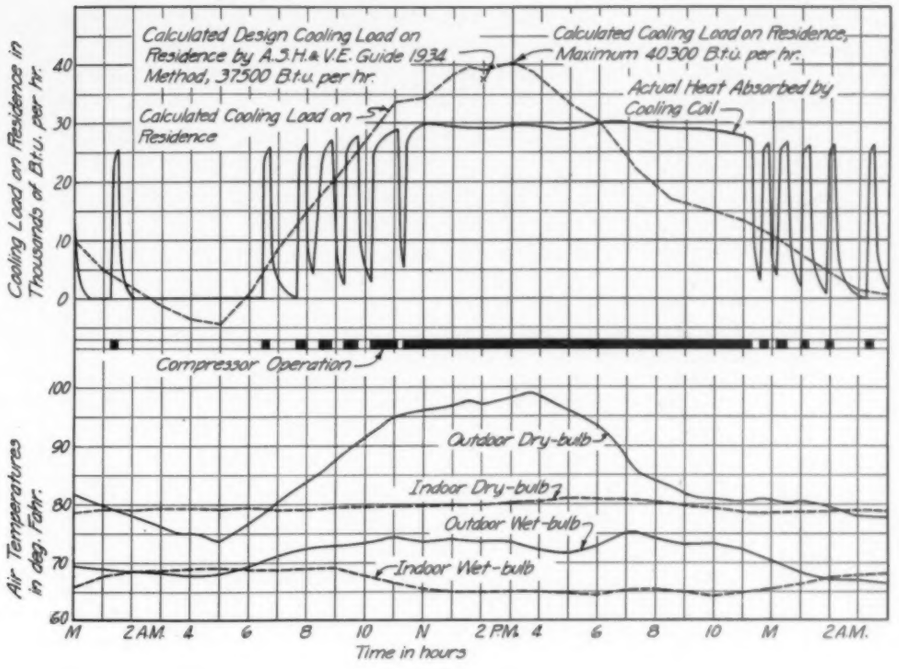

Calculated Design Cooling Load on Residence by A.S.H.& V.E. Guide 1934 Method, 37500 Btu per hr.
Calculated Cooling Load on Residence, Maximum 40300 Btu per hr.
Actual Heat Absorbed by Cooling Coil
Calculated Cooling Load on Residence
Cooling Load on Residence in Thousands of Btu. per hr.
Compressor Operation
Outdoor Dry-bulb
Indoor Dry-bulb
Outdoor Wet-bulb
Indoor Wet-bulb
Air Temperatures in deg. Fahr.
M 2AM 4 6 8 10 N 2PM 4 6 8 10 M 2AM
Time in hours

Fig. 6—Actual and calculated cooling load on Residence and air temperatures, Test No. 7-34, June 27, 1934; no night air cooling

dows was obtained by taking 28 per cent of the heat calculated for unshaded windows, as indicated by more recent studies of F. C. Houghten[5] and others.

The heat transmitted through the walls due to the action of the sun on the exposed portions was calculated by the following method. Of the total intensity of the sun a part, I, depending on the angle of incidence, is received by the surface. A portion, eI, depending on the absorption coefficient, e, is absorbed by the surface. This tends to raise the temperature of the surface and of, eI, a portion, H_1, is lost by radiation and convection to the outdoor air, and another portion, H_2, is transmitted through the wall to the indoor air. From these considerations the following equations may be established[6]:

$$eI = H_1 + H_2, \text{ or } H_2 = eI - H_1$$
$$H_2 = (t_X - t_Y)\, C = (t_Y - t_i)\, f_i$$
$$H_1 = (t_X - t_o)\, f_o$$

In which t_o = temperature of outdoor air, degrees Fahrenheit.
t_i = temperature of indoor air, degrees Fahrenheit.
t_X = temperature of outside surface of wall, degrees Fahrenheit.
t_Y = temperature of inside surface of wall, degrees Fahrenheit.
I = Sun intensity for a given angle of incidence, Btu per square foot per hour.
e = emissivity, or absorption coefficient.
H_1 = portion of eI lost to outdoor air, Btu per square foot per hour.
H_2 = portion of eI transmitted through wall, Btu per square foot per hour.
f_o = outside surface coefficient, Btu per square foot per hour per degree difference in temperature between t_X and t_o.
f_i = inside surface coefficient, Btu per square foot per hour per degree difference in temperature between t_X and t_i.
C = conductance of wall, Btu per square foot per hour per degree difference in temperature between t_X and t_Y.

For given indoor and outdoor temperatures and a given sun intensity the portion, H_2, of the heat transmitted through the wall may be obtained by the simultaneous solution of these equations. For the purpose of calculating the load for the Research Residence a value of 0.40 was assumed for the absorption coefficient, e, and it was assumed that when the sun was shining on a wall 50 per cent of the net area was actually exposed to the sun and 50 per cent was shaded by shadows from the awnings, shutters, etc. The latter assumption was based on studies of the amount of shading at different times of the day.

For the purpose of comparison the design load calculated by the method outlined in THE A.S.H.V.E. GUIDE 1934 has been shown as a single point in Fig. 6. Certain modifications and assumptions were necessary in applying this method. The heat from the sun transmitted through the windows was obtained by using a value taken at 2 p.m. from the intensity curves given in THE GUIDE, and by multiplying this result by 0.83[2] to allow for heat intercepted by the glass and by 0.28[5] to allow for the shading effect of the awnings. In order to obtain the sun effect on the walls a value of 25 F was added to the difference in temperature between outdoor and indoor air as recommended. The south and west walls only were regarded as exposed to the sun, and 50 per cent of the net area of these walls was regarded as shaded by awnings and shutters. The design temperature of 91 F was selected from the table in THE GUIDE and the average of 80 F was used for the indoor temperature. The temperature in the attic was assumed as 15 F higher than the outdoor temperature. It may be observed that this calculated design load was 37,500 Btu per hour as compared with the actual load of approximately 30,000 Btu per hour. The latter proved sufficient to cool the house satisfactorily.

[5] A. S. H. V. E. Research Paper entitled, "Studies of Radiation Through Bare and Shaded Windows," by F. C. Houghten, C. Gutberlet, and J. L. Blackshaw. (*Heating, Piping and Air Conditioning*, February, 1934, pp. 67-74.)

[6] "Mechanical Equipment of Buildings," Vol. 1, second edition, by L. A. Harding and A. C. Willard, p. 248.

Table 2—Complete Data and Results for Tests With Mechanical Refrigeration

Date 1934	Test No.	Series No.	WEATHER DATA: Degree Hrs. Above 85°	WEATHER DATA: Degree Hrs. Above 90°	WEATHER DATA: Sun Hr. Min.	WEATHER DATA: Outdoor Air: Max. Temp. During Test, F	WEATHER DATA: Outdoor Air: Min. Temp. Previous to Test, F	WEATHER DATA: Outdoor Air: Average During Period Requiring Mechanical Refrig. Temp. F	WEATHER DATA: Outdoor Air: Average During Period Requiring Mechanical Refrig. Hum. %	Indoor Air Conditions: At End of Night Cooling Temp. F	Indoor Air Conditions: At End of Night Cooling Hum. %	Indoor Air Conditions: Average During Period Requiring Mechanical Refrig. Temp. F	Indoor Air Conditions: Average During Period Requiring Mechanical Refrig. Hum. %	Temperature Difference: Outdoor Max. to Avg. Indoor F	Temperature Difference: Average Outdoor to Avg. Indoor F	Night Air Cooling: Previous to Test: Duration in Hrs.	Night Air Cooling: Previous to Test: Power Input to Fan Kw Hr	Overall Test Period: Start	Overall Test Period: End	Overall Test Period: Duration Hr	[a]Circulating Fan, Power Input During Overall Test Period Kw Hr	Compressor Operation: Time in Hrs.	Compressor Operation: Power Input, Kw Hr	Compressor Operation: Condenser Water Gal	Heat Absorbed by Cooling Coil During Overall Test Period, Btu: Total Moisture Condensed Lb.	Heat Absorbed by Cooling Coil During Overall Test Period, Btu: Load Due to Moisture	Heat Absorbed by Cooling Coil During Overall Test Period, Btu: Sensible	Heat Absorbed by Cooling Coil During Overall Test Period, Btu: Total	Total Heat Absorbed by Cooling Water During Overall Test Period, Inlet to Drain, Btu	Heat Equiv. of Total Power Input to Compressor During Overall Test Period Btu	Total Net Heat Gain on House During Overall Test Period Btu	Ratio of Net Heat Gain on House to Heat Absorbed by Cooling Coil
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33
June 21	2-34	4-34	21.00	0	15:06	89.5	70.0	86.7	41.4	.....	.....	79.6	46.4	9.9	7.1	13.00	.....	9:00A	9:00A	24.00	5.3	6.67	18.9	514	41.27	43,333	136,444	179,777	188,181	64,534	123,647	0.688
23	3-34	3-34	39.50	8.00	15:05	93.0	66.5	86.8	43.8	74.8	80.5	79.7	50.2	13.3	7.1	8.25	3.6	2:32P	10:45P	8.22	2.2	5.73	16.4	482	37.33	39,197	131,728	170,925	174,700	58,998	118,702	0.694
24	4-34	3-34	0	0	13:30	84.5	65.9	82.9	66.1	74.8	79.0	79.2	69.0	5.3	3.7	8.25	2.3	5:04P	7:35P	2.52	0.6	0.83	2.3	.....	0.98	7,330	.....	.....	.....	7,853	.....	.....
25	5-34	3-34	25.00	3.75	12:20	93.9	69.4	85.6	55.3	76.1	79.5	80.4	60.8	14.4	6.1	11.42	5.1	12:55P	9:38P	8.67	2.8	5.00	14.6	486	44.67	46,904	106,129	153,033	168,900	49,852	119,048	0.778
26	6-34	3, 4-34[b]	84.50	36.00	11:59	97.7	75.6	85.8	45.9	79.6	55.6	80.1	50.2	18.1	6.2	9.42	4.2	11:17A	10:00A	22.72	7.4	11.92	34.5	1115	87.85	92,242	237,000	329,242	370,900	117,800	253,100	0.769
	7-34	4-34	104.00	53.50	15:04	99.0	73.8	86.9	49.6	79.6	51.5	79.6	51.5	19.4	7.3	.....	.....	10:00A	9:00A	23.00	7.9	15.33	44.4	1519	116.14	121,947	340,868	462,815	505,100	151,604	353,496	0.764
28	8-34	4-34	127.00	63.25	15:03	99.9	75.8	89.0	48.2	.....	.....	79.6	47.1	20.3	9.4	.....	.....	9:00A	9:00A	24.00	8.0	19.87	56.6	1890	137.60	144,490	427,530	572,010	633,100	193,261	439,839	0.769
29	9-34	4, 3-34[c]	70.25	27.25	9:59	97.5	77.7	89.1	50.4	79.6	47.0	79.7	47.0	17.9	9.5	.....	.....	9:00A	10:30P	13.50	4.4	12.37	35.5	1180	95.87	100,663	261,237	361,900	396,600	121,215	275,385	0.761
30	10-34	3-34	0.50	0	8:44	86.2	71.3	82.0	58.1	78.5	69.5	79.7	57.9	6.5	2.3	8.50	3.8	11:00A	7:00P	8.00	2.6	2.20	6.2	213	11.48	12,054	51,674	63,728	70,100	21,170	48,930	0.768
July 1	11-34	3-34	36.25	3.50	15:03	92.0	67.7	87.8	34.2	75.9	67.5	80.1	50.2	11.9	7.7	12.00	5.4	2:57P	8:20P	5.38	1.8	3.25	9.4	302	15.73	16,516	76,832	93,348	102,700	33,096	70,604	0.756
	12-34	3-34	55.25	17.00	15:03	96.0	65.8	89.2	37.8	75.6	56.5	80.2	45.2	15.8	9.0	10.67	4.8	2:20P	8:45P	6.42	2.1	4.83	13.8	432	24.92	26,166	107,588	133,754	147,000	47,120	99,880	0.747
3	13-34	3-34	66.00	19.75	14:32	95.0	72.6	90.3	31.2	77.5	54.5	80.7	45.9	14.3	9.6	10.25	4.6	11:40A	8:50P	9.17	3.0	4.78	13.5	427	16.15	16,988	117,658	134,616	148,800	46,096	102,704	0.763
4	14-34	3-34	54.75	14.25	10:30	94.7	71.3	89.2	40.2	76.6	47.0	80.6	49.6	14.1	8.6	10.17	4.6	1:58P	10:00P	8.03	2.6	4.22	12.0	376	17.78	18,670	98,923	117,593	129,700	40,974	88,726	0.755
5	15-34	3, 4-34	84.50	31.50	9:00	97.0	75.2	86.9	53.2	78.5	66.5	80.4	54.7	16.6	6.5	9.00	4.0	11:00A	9:00A	22.00	7.3	13.33	38.2	1206	80.60	94,080	288,005	382,085	414,200	130,434	283,766	0.743
6	16-34	4-34	3.75	0	5:25	87.9	68.6	84.1	52.1	.....	.....	80.4	55.0	7.5	3.7	.....	.....	9:00A	7:30A	22.50	7.5	4.00	11.3	355	29.22	27,530	82,840	110,370	121,900	38,584	83,316	0.755
	17-34	3-34Grav.	36.75	6.50	14:46	93.7	67.2	85.5	49.3	75.1	62.0	80.2	55.3	13.5	5.3	11.50	5.2	4:36P	10:00P	5.40	1.8	2.18	6.4	172	11.58	12,160	42,330	54,490	59,730	21,833	37,877	0.695
10	18-34	3-34	53.25	11.25	12:14	93.1	75.5	87.6	57.8	77.1	82.5	80.0	59.7	13.1	7.6	8.75	4.6	1:08P	10:15P	9.12	3.0	7.68	22.1	535	63.87	67,064	115,388	182,452	202,700	75,460	127,240	0.697
15	19-34	3-34	37.75	3.50	14:44	92.4	70.0	87.0	59.8	76.1	76.5	80.4	63.0	12.9	6.6	11.33	5.2	1:57P	10:30P	8.55	2.9	7.10	19.9	453	55.39	58,160	94,810	152,970	169,900	67,949	101,951	0.666
18	20-34	3-34	23.25	0	10:57	89.5	71.0	85.7	69.7	76.3	74.5	80.7	65.5	8.8	5.0	8.50	3.9	2:30P	10:45P	8.25	2.7	7.20	20.2	422	56.81	59,680	93,000	152,680	165,100	68,973	96,127	0.630
19	21-34	3, 4-34	96.25	42.25	14:41	97.8	76.7	88.0	58.6	79.4	74.5	81.2	67.2	16.6	6.8	8.25	3.7	9:00A	9:15P	12.25	4.1	21.82	61.0	1170	153.65	161,333	293,271	454,604	464,600	208,285	256,315	0.564
	22-34	4-34	131.75	73.75	14:40	102.0	78.2	88.2	64.2	82.2	56.7	19.8	6.0	.....	.....	9:00A	9:00A	24.00	8.0	24.00	66.5	150.12	157,626	300,156	457,782	471,900	227,064	244,836	0.535
22	23-34	4-34	107.50	53.75	14:38	100.0	77.2	87.0	61.7	82.6	55.6	17.4	4.4	.....	.....	9:00A	9:00A	24.00	7.7	24.00	65.9	119.69	125,675	278,109	403,784	414,000	225,016	188,984	0.468
23	24-34	4-34	141.25	85.25	14:36	102.8	72.8	87.8	55.2	83.5	53.0	19.3	4.3	.....	.....	9:00A	9:00A	24.00	7.8	24.00	64.6	89.19	93,650	257,153	350,803	348,200	220,577	127,623	0.364
25	26-34	3, 4-34	117.00	57.25	11:52	101.4	76.9	87.8	51.9	80.3	48.7	21.1	7.5	.....	.....	9:00A	9:00A	24.00	7.8	18.00	51.7	124.80	131,135	391,933	523,068	562,000	170,530	385,470	0.737
26	27-34	4, 3-34	49.50	8.25	8:25	94.1	71.5	88.3	51.6	80.2	50.4	13.9	8.1	.....	.....	9:00A	9:15P	12.25	4.1	9.38	26.9	902	67.71	71,096	203,062	274,158	298,700	91,850	206,850	0.754
	28-34	3-34	13.50	0.25	10:33	90.8	69.7	84.2	70.3	76.3	80.5	80.8	74.5	10.0	5.8	13.00	5.8	4:50P	6:50P	2.00	0.7	0.50	1.5	48	4.03	4,230	11,027	15,257	16,400	5,122	11,278	0.739
Aug. 2	29-34	4-34	45.75	11.00	10:19	93.0	73.2	83.1	83.5	.....	.....	80.0	71.8	13.0	3.1	.....	.....	9:00A	9:00A	24.00	8.0	8.00	23.5	791	94.56	99,288	157,302	256,590	276,800	80,241	196,559	0.766
7	30-34	4-34	59.25	12.50	14:04	94.8	76.2	85.5	76.4	.....	.....	80.0	67.1	14.8	5.5	.....	.....	9:00A	9:00A	24.00	8.0	13.87	40.6	1340	172.82	181,460	262,500	443,960	480,500	138,629	341,871	0.770
8	31-34	4-34	94.75	47.00	14:02	99.2	79.2	87.5	71.1	.....	.....	80.5	56.9	18.7	7.0	.....	.....	9:00A	8:00A	24.00	8.0	16.12	46.4	1369	141.49	148,565	315,235	463,800	501,500	158,433	343,067	0.740
9	32-34	4-34	62.75	20.25	14:00	94.2	68.3	90.6	56.0	.....	.....	80.2	54.8	14.0	10.4	.....	.....	9:00A	9:00A	24.00	7.9	9.83	28.2	872	85.42	89,690	203,587	293,277	315,800	90,280	219,511	0.748

Tests Nos. 1, 25, 33 and 34 were omitted on account of incomplete data and unfavorable changes in the weather. [b]3, 4 is a series No. 3-34 test. [c]4, 3-34 is a series No. 4-34 test. [d]Circulating fan was run continuously during the overall test period.

The indoor relative humidity was approximately 61 per cent until 9 a.m., as indicated in Fig. 6. During the early period of intermittent operation of the compressor, the relative humidity decreased until the compressor operated continuously, at which time it assumed a value of approximately 45 per cent and remained practically constant at this value until the compressor again began to operate intermittently.

Characteristic results obtained during the continuous operating periods of the compressor are shown in Table 1. The air conditions at different locations are given in the first ten items of the table. The dry-bulb temperature on the second story was only 1.0 F higher than that on the first, thus representing a very satisfactory balance of the cooling on the two stories. The ventilating air was taken from a large sunken window areaway on the north side of the Residence. This areaway was at the junction of the main body of the house and the north wing forming the kitchen, and hence was protected from the sun on both the south and the west. The temperature of the ventilating air was therefore somewhat lower and the relative humidity somewhat higher than the corresponding values for the outdoor air. The temperature of the cooled air leaving the registers was never lower than 68.0 F and no difficulty was experienced from drafts in the rooms. The rise in temperature of the air passing through the furnace casing and ducts was 4.4 F. The quantity of air circulated was 1,303 cfm, which amounted to 5.5 recirculations per hour or 540 cfm per ton of refrigeration absorbed from the air. With this amount of air the velocity through the face area of the coil was 368 fpm and the temperature drop was 15.9 F. This corresponded closely with the 350 fpm and the 15 F drop used for the design. The free area velocity of 598 fpm did not prove to be sufficient to carry any condensed

moisture away from the surfaces of the cooling coil.

The useful refrigerating effect, or heat absorbed by the cooling coil was 29,065 Btu per hour, which compared favorably with the nominal rating of 29,500 Btu per hour, although the temperature of the refrigerant as shown by Item 25 was considerably above the 32 F on which the rating was based. Item 25 indicates that considerable superheating occurred in a large portion of the evaporator or cooling coil. The ratio of the useful refrigerating effect to the heat absorbed by the water passing through the condenser was 0.95. This high ratio may be explained by the fact that considerable heat was lost from the condenser coil to the air in the basement, which would otherwise have been absorbed by the water and thus would have reduced the value of the ratio. However, this ratio, obtained for long operating periods of the compressor on each test, proved valuable in checking the calculated refrigerating effect for the periods of intermittent operation, during which the validity of using average temperatures for the air and average weights for the water condensed on the coil is somewhat uncertain. Such calculations based on averages for widely fluctuating conditions resulting from intermittent operation are always subject to some uncertainty.

As shown in Item 16 the useful refrigerating effect, or heat absorbed by the cooling coil, was obtained by computing the heat absorbed by the air, which was separated into the sensible heat given up by the dry air and the heat given up by the change in the moisture content of the air in passing through the cooling coil. In connection with the latter heat quantity, some difficulty was experienced in obtaining accurate readings of the wet-bulb temperatures as observed by means of recording thermocouples. The wicks used on these recording thermocouples became fouled in a short time, thus affecting the accuracy of the observed wet-bulb temperature. However, a sufficient number of accurate readings was obtained to prove that no great amount of variation occurred in the weight of air and in the entering and leaving wet- and dry-bulb temperatures for the air passing through the cooling coil during periods of compressor operation. The weighed amounts of water condensed on the coil were therefore accepted as being accurate, and the heat absorbed was obtained by multiplying this weight by a constant value representing the heat given by the change in moisture content of the air per pound of water vapor condensed. This latter included the latent heat and superheat in the water vapor condensed, and the change in the superheat in the water vapor remaining in the air after passing the cooling coil, and amounted to 1,050 Btu per pound of vapor condensed. The weight of water vapor condensed varied from 12.5 to 3.4 pounds per hour based on the periods of continuous operation for the compressor, or from 7.2 to 1.2 based on the total length of the test period over which artificial cooling was required.

Item 21, the heat absorbed by the water passing through the condenser, the compressor heads, and the piping in the basement, H_w, represents all of the heat that was removed from the house with the exception of that which might have been lost through the basement walls. In order to separate the heat gains introduced by the presence of the refrigerating machine in the basement, such as electrical and mechanical losses in the compressor motor, mechanical losses in the compressor and drive, and heat losses from the condenser and compressor heads, from the net heat gains which were independent of the presence of the machine, the following heat balance on the house as a whole may be made:

$$H_w = H + H_e, \text{ or } H = H_w - H_e$$

In which H_w = Total heat removed by the cooling water from the point of entrance to the point of drainage.

H = Net heat gain of house, including heat transmission through walls, ceilings, and glass, infiltration, ventilating air, heat from lights and fan motor, and heat from occupants, but excluding heat lost through the basement walls.

H_e = Heat equivalent of electrical input to the compressor motor.

The net heat gain, H, is shown as Item 23 in Table 1. Item 16, the heat absorbed by the cooling coil, represents the useful refrigerating effect, and since ultimately all of the cooling in the house had to be accomplished by means of the air passing through the coil, it also represents the total amount of refrigeration that had to be expended to cool the house under the conditions of operation with the machine in the basement. If this heat is represented by H_r, then the ratio $\frac{H}{H_r}$ becomes an index of the additional cooling load imposed by the presence of the refrigerating machine in the basement, since this ratio would be 1.0 if it were possible to operate with no heat loss from the machine or condenser. In some respects this ratio is analogous to the overall house efficiency which may be obtained for winter heating. The value of 0.767, given as Item 24, is relatively low as compared with overall house efficiencies obtained for winter heating. Since $H = H_w - H_e$ is the net heat gain for the house, it may be observed that for given outdoor weather conditions any increase in the heat loss from the machine or condenser must be compensated for by readjustments in H_w and H_e. These readjustments must be such that the difference, H, remains constant. Any such increase in losses, however, will be directly reflected in an increase in H_r. Hence the ratio $\frac{H}{H_r}$ will decrease rapidly as the losses from the machine or condenser are increased. Insofar as operating costs are concerned, it would therefore seem to be advantageous to insulate the condenser or to install the machine outside of the house.

The general results for all of the tests with mechanical refrigeration are shown in Table 2. It may be observed from Column 33 that beginning with test No. 17-34, a decrease in the ratio $\frac{H}{H_r}$ occurred up to, and including, test No. 24-34. This was caused by a gradual loss of the charge of refrigerant occurring over this period. As a result, the compressor did not operate as efficiently, and the heat losses from the machine and condenser became a larger portion of the useful refrigerating effect. The refrigerant had no odor and the gradual loss of the charge was obscured by other factors. A small decrease in the drop in temperature was observed, but it was erroneously attributed to an increase in outdoor relative humidity which occurred at about

this time. Furthermore, the electrical input to the compressor motor which was observed by means of an integrating wattmeter did not decrease sufficiently to attract attention. The indoor temperature, however, was maintained lower than 81 F until test No. 22-34. Hence, only the three tests, Nos. 22-34 to 24-34, inclusive, have been regarded as of doubtful value, and these are useful in determining the effect on comfort of maintaining an indoor temperature higher than 81 F, which is discussed later. The machine was recharged after test No. 24-34 and on test No. 26-34, during which the maximum outdoor temperature rose to 101.4 F, the ratio $\frac{H}{H_r}$ rose to a normal value and the indoor temperature was maintained at 80.3 F, indicating that with a normal charge of refrigerant no difficulty would have been encountered in maintaining an indoor temperature of 80 F on the three preceding days for which the maximum outdoor temperature was from 100.0 F to 102.8 F. This experience, however, in which the loss of the charge was obscured by other factors even with the machine under constant observation, indicates that from the standpoint of the householder, it would be advantageous to introduce some odor, or other positive means of indicating a leak before the loss of the whole charge has occurred.

The total cooling load for the day, expressed as the total Btu absorbed from the air during the overall period of compressor operation, is shown in Fig. 7 plotted against the degree hours above 85 F per day. The base temperature of 85 F was selected because it was found that usually no cooling was required unless the outdoor temperature was approximately 85 F or above. Curves for both total heat and sensible heat are shown, and the difference between these curves represents the heat due to the change in the moisture content of the air. This moisture load varied from approximately 25 per cent to 30 per cent of the total load. One point, at 60 deg hours, deviates widely from the total heat curve but does not deviate very widely from the sensible heat curve. On this particular test, No. 29-34, the outdoor relative humidity was 83.5 per cent, the highest recorded during the summer, and the load due to moisture was excessive.

A comparison between the results from *Series 4-34*, for which no cooling with outdoor air at night was employed, and those from *Series 3-34*, for which supplementary night cooling was used, may be obtained from Fig. 7. The data for tests Nos. 22-34 to 24-34 inclusive have been omitted from these curves because the average indoor temperature was higher than 81 F and these tests were not regarded as being comparable with the others. The results from the two series of tests form two fairly well defined curves. More deviations occurred in the points representing the data from *Series 3-34* because the amount of artificial cooling required during the day was directly influenced by the outdoor conditions on the preceding night. The curve represents median conditions, and if the night was exceptionally cool less artificial cooling was required the next day, while if the night was exceptionally warm the reverse was true. In the case of *Series 4-34*, for which the windows were not opened at night, the variations in the outdoor conditions at night were not so directly reflected in the amount of cooling required and hence less deviations occurred between the median curve and the points representing individual tests.

From the curves in Fig. 7 it is apparent that considerable saving in the amount of mechanical refrigeration required for cooling may be effected by supplementary cooling with outdoor air at night, even when the latter is confined to that which may be obtained by opening only the second story windows at night and employing 9.5 air changes per hour as furnished by the fan in the

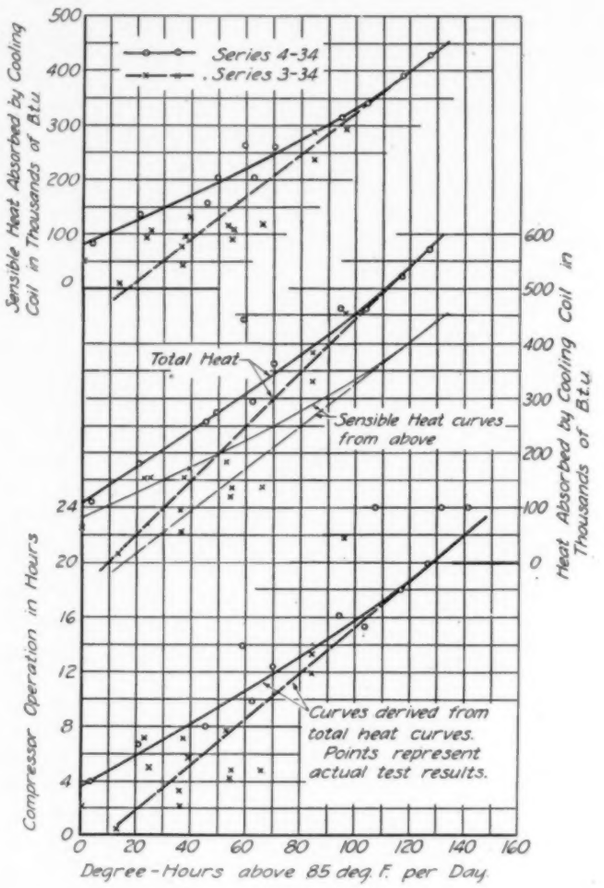


Fig. 7—Heat absorbed by cooling coils and hours of compressor operation per day

forced-air heating system. This is particularly true for milder weather. For *Series 3-34*, the windows were opened when the outdoor effective temperature became the same as that indoors. As the nights became warmer, this opening time occurred later, and when the degree hours per day reached a value of approximately 110 no windows were opened and *Series 3-34* merged into *Series 4-34*. On days for which the maximum outdoor temperature reached 80 to 85 F the windows were opened at about 7 p. m. and no cooling was required the next day. When the maximum outdoor temperature was from 85 to 95 F the windows were opened sometime between 8 and 10 p.m. and artificial cooling was required the next day. When the maximum outdoor temperature was above 95 F and the indoor and outdoor effective temperatures did not equalize until later than midnight, the

windows were not opened and artificial cooling was employed during the whole of the 24 hours.

The curves in the lower portion of Fig. 7 were computed from the total heat curves in the upper portion, and represent the total time that the compressor would be required to run delivering a constant mean refrigerating load of 29,000 Btu per hour. The actual running times obtained from the observed data for the individual tests have been plotted as points on the curves. A study of these points shows that, with the exception of four points near the upper end of the curves, they fell in about the same positions relative to the curves as the points fell relative to the total and sensible heat curves in the upper portion of Fig. 7. Of the exceptions, the three upper points represent data from tests Nos. 22-34 to 24-34 inclusive, which were excluded from the heat curves because an indoor temperature below 81 F was not maintained. The fourth point represents test No. 21-34, which was included in the upper curves because the indoor temperature was maintained at 80.7 F. From Column 33 in Table 2 it may be noted that for these four tests exceptionally low values of the ratio $\frac{H}{H_r}$ were obtained. A study of the points therefore indicates that when the refrigerant leak started with test No. 17-34, a gradual loss in efficiency occurred as evidenced by the reduction in the ratio of $\frac{H}{H_r}$, but the loss of part of the charge did not affect the output of the machine until test No. 21-34 was reached. At this point the output was decreased and although it was still possible to maintain an average indoor temperature below 81 F it was accomplished by an increase in the running time of the compressor. For the points representing tests Nos. 22-34 to 24-34 inclusive, it was not possible to maintain a temperature below 81 F even with the compressor operating continuously for 24 hours. Hence these three tests have been excluded in any consideration of the data. The point at 60 deg hours, representing test No. 29-34, is of interest as indicating the increase in running time that was made necessary by the moisture load imposed by an outdoor relative humidity of 83.5 per cent. It may also be observed that extrapolation of the curves to the 24 hour ordinate indicates that the compressor would not be required to operate continuously over the 24 hour period until the number of degree hours per day above 85 F equalled, or exceeded, approximately 150. Since this condition does not occur at Urbana, Ill., the data indicate that a refrigerating machine of 2½ tons capacity is ample to maintain an indoor temperature not in excess of 81 F in a house similar to the Research Residence located where the climate is similar to that at Urbana, Ill.

On all of the tests in which the average indoor temperature for the test was not above 81 F the maximum indoor temperature did not rise above 81.5 F, and then only for a short time. With indoor relative humidities between 45 and 55 per cent this was equivalent to effective temperatures between 74 and 75 F. During the daytime these conditions were found to be comfortable both by members of the staff who occupied the Residence more or less constantly and by visitors who were present for short periods only. The maximum discomfort occurred with the *Series 3-34* tests during the hour just preceding the start of the compressor. After the windows were closed in the morning, the indoor dry-bulb temperature did not rise sufficiently to start the compressor until the late morning or early afternoon hours. During this time that the house was closed and the plant was not operating the indoor relative humidity increased and for a short time the house felt stuffy. Odors were also somewhat noticeable at this time. When the plant was started, both the slight discomfort and the odors disappeared. During the *Series 4-34* tests, for which the windows were not opened, the plant operated either continuously or at short intermittent periods over the whole 24 hours, and no discomfort or odors were noticeable. The admission of outdoor air for ventilation amounting to approximately one air change per hour was effective in overcoming any odors. At night, a temperature of 80 F was not entirely comfortable for a person lying down. The bed prevented the circulation of air over part of the body and the remaining surface was apparently not sufficient for effective cooling. It is therefore possible that sleeping quarters should be cooled to a somewhat lower temperature than living quarters.

Table 3—Summary of Results of Tests with Mechanical Refrigeration and Night Air Cooling for Entire Season of 1934.

1.	Total hours above 85 F for season of 1934	481.5
2.	Total hours above 90 F for season of 1934	224.1
3.	Total degree-hours above 85 F for season of 1934	2,657.5
4.	Total degree-hours above 90 F for season of 1934	948.75
5.	Number of tests with mechanical refrigeration	33
6.	Total running time for fan during night cooling, hours	692.3
7.	Average rate of power input to fan during night cooling, watts	410
8.	Total power input to fan during night cooling, Kwhr	283.8
9.	Total running time for fan during test period, hours	559.2
10.	Average rate of power input to fan during test period, watts	314
11.	Total power input to fan during test period, Kwhr	175.5
12.	Total running time for fan including night cooling, hours	1,251.5
13.	Total power input to fan including night cooling Kwhr	459.3
14.	Total running time for compressor, hours	335.7
15.	Average rate of power input to compressor, watts	2,836
16.	Total power input to compressor, Kwhr	952.1
17.	Total power input to compressor and fan, including night cooling, Kwhr	1,411.4
18.	Total quantity of cooling water, gallons	25,741
19.	Total heat absorbed by cooling coil during season, Btu	8,354,881
20.	Equivalent ice meltage during season, tons	29.0
21.	Corrected ice meltage during season assuming water to the drain of 40 F, tons	27.5

Table 3 gives a summary of total quantities obtained for the season of 1934. The total of 2,657.5 deg hours above 85 F, Item 3, indicates that this season was severe as compared with those for 1933 and 1932, for which the degree hours above 85 F were 2,309.3 and 1,470.7 respectively. This was caused by the comparatively large number of days on which the outdoor temperature rose above 90 F. The hours above 85 F for the seasons of 1934, 1933, and 1932 were 481.5, 493.2, and 329.1 re-

spectively, while those above 90 F were 224.1, 208.4, and 121.8.

It is of interest to note that the total heat absorbed during the season with the two methods of operating the Residence from June 20 to September 16 was 8,354,881 Btu, which was equivalent to an ice meltage of 29.0 tons. It is not usual to reject the water to the drain at a temperature of 32 F in the case of an ice plant. On the assumption that the water is rejected at approximately 40 F and that advantage is taken of this additional cooling effect, an equivalent ice meltage of 27.5 tons would be obtained as shown by Item 21. This equivalent tonnage would have been slightly increased if the plant had been in operation about May 15. However, the few hot days in the latter part of May and the early part of June were accompanied by cool nights, and by employing night cooling with outdoor air, it would not have been necessary to start the plant during this time with the exception of not more than five days. The 27.5 tons may, therefore, be regarded as representative of the whole season's requirements under the existing conditions of operation. This figure is *not* comparable with the 43.3 tons of ice actually used during the summer of 1932. For the 1932 season the windows remained closed during the whole period from June 1 to October 1, and no advantage was taken of the possible supplementary cooling with outdoor air at night. Furthermore, for one-half of the season no awnings were used on the windows. For the work in 1934, full advantage was taken of cooling with outdoor air at night for a large part of the season and the Residence was equipped with awnings during the whole season. The 27.5 tons is to a certain extent comparable to the estimated 19 tons for the season of 1933 obtained under conditions of night cooling with the fan in the forced-air heating system and windows on both stories opened; if it is recognized that the season of 1934 was more severe than that of 1933, that the maximum amount of ice used per day was limited to 700 lb during the season of 1933, and that night cooling with the windows opened on both stories during the season of 1933 was somewhat more effective than similar night cooling with second story windows only opened during the season of 1934.

A discussion of comparative costs is not conclusive unless unavoidable differences in existing conditions are fully recognized. However, the total cost for electricity and water for the season's operation under local conditions, is of some interest. The total electrical input to the compressor and fan, including the input to the fan during night cooling, as shown by Item 17, was 1411.4 kwhr. Therefore at the local rate of 0.031 per kwhr the cost for electrical current was $43.75. The total amount of water used, as shown by Item 18, was 25,741 gal. At the local rate of $0.37 per thousand gallons the total cost for water was $9.52. Hence, the total cost for electricity and water under the conditions of operation was $53.27. As previously mentioned, if the cooling plant had been operated from May 15 to September 16 instead of from June 20 to September 16, this figure would have to be slightly increased. However, the season would not have been increased by more than five days on which cooling was necessary and the plant would have had to operate during only a small portion of each day. On the other hand, if supplementary cooling with outdoor air had been used during the whole season a reduction in this cost would have been probable. Therefore, it seems reasonable to regard $60.00 as the approximate cost for electricity and water for the season of 1934.

Conclusions

The following conclusions may be drawn as applying to the Research Residence and the conditions under which the tests were conducted:

(1) An indoor temperature of approximately 80 F with relative humidity below 55 per cent results in satisfactory comfort conditions in the living quarters of a residence. For complete comfort in sleeping quarters a somewhat lower temperature is desirable.

(2) The introduction of approximately one air change per hour of outdoor air for the purpose of ventilation is sufficient to prevent objectionable odors.

(3) A mechanical refrigeration unit capable of producing 2½ tons of refrigeration is sufficient to maintain conditions of comfort on two stories of a residence similar to the Research Residence when the outdoor temperature does not exceed 103 F, and an amount of outdoor air equivalent to one air change per hour is used for the purpose of ventilation.

(4) The use of the fan in the forced-air heating system to circulate outdoor air at night amounting to 9.5 air changes per hour is not as effective in cooling the residence as a whole when the second story windows only are opened as it is when windows on both stories are opened.

(5) The cooling load during the daytime can be reduced by supplementing artificial cooling during the day with cooling by means of outdoor air circulated at night.

(6) Satisfactory cooling with outdoor air at night probably cannot be accomplished by employing less than 9 air changes per hour. The gain arising from the use of more than 30 air changes per hour is very small.

Heating a Building with Cold Water

Air conditioning equipment of the reversible-cycle refrigerating type, adaptable to either heating or cooling, depending on the season, is now in operation in the new building of the Atlantic City Electric Co. at Salem, N. J. The building is a two-story and basement, brick and steel reinforced concrete structure, insulated, and of modern design.

Installations of a similar type have been made previously and in the majority of cases have depended on outdoor air as the source of heat. In the Atlantic City Electric Co. building, the heat is drawn from a well of water, which maintains a natural temperature of at least 56 deg in the coldest weather. The heat is transferred from the water to the refrigerant in a large water cooler, with the water leaving the cooler at about 40 deg. In the compression cycle, the temperature of the refrigerant is raised to 135 deg and it gives up its heat to the air within the building by passing through a condenser, over which air is circulating.

In summer, this condenser will serve as a cooling surface to cool and dehumidify the air and the heat thus absorbed will be dissipated by the water cooler, acting as a water heater. Humidifiers for winter use, air filters, and a high velocity fan and air circulating system complete the air conditioning apparatus, maintaining close automatic control of both temperature and humidity within the building in summer and winter.

The Thermo-Integrator—a New Instrument for the Observation of Thermal Interchanges

By C.-E. A. Winslow[1] and Leonard Greenburg,[2] New Haven, Conn.

(MEMBERS)

THERE are four distinct external factors which may govern the interchange of heat between the human body and its environment—the temperature of the air and the velocity of movement of the air (governing both convection and evaporation), the relative humidity of the air (governing evaporation), and the temperature of surrounding surfaces (governing radiation). Students of air conditioning have for over a century been seeking, and are still seeking, for the simplest and most accurate method of measuring these four factors and their combined effect upon human comfort.

The classic instrument in use is the thermometer but, while this instrument registers one of the four factors accurately, it gives no measure of either air movement or relative humidity and is affected only in slight and variable degree by radiation. The wet bulb thermometer brings a second factor, relative humidity, within the scope of observation and the Radif thermometer of Lindsay (1933)[3] gives us some measure of radiation, although a very imperfect one, on account of the size and construction of the instrument.

Over a century ago Heberden (1826)[4] realized the importance of recording the influence of air movement and introduced a new principle into the practice of thermometry. He heated his thermometer to above 120 F and observed the time taken by the mercury level to fall for a given distance, under the cooling influence of the air. This is, of course, the principle later developed by Hill (1913),[5] Hill, Griffith and Flack (1916),[6] Hill (1919),[7] Hill (1923 *a* and *b*),[8,9] in the construction of the Kata-thermometer, which has proved an admirable instrument for measuring the two factors of air temperature and air movement. By adding a wet bulb the factor of relative humidity was also brought into the picture, and Hill believed that his wet Kata represented better than any other instrument the combined influence of all atmospheric factors upon the human body. Later studies have shown, however, that this is only even approximately true at certain temperatures and the range of the wet Kata is so narrow that it has not proved of very great value in practical ventilation studies.

About 10 years ago a great deal of attention was devoted to the problem of measuring the combined influence of air temperature, air movement and relative humidity by methods which depended on the same principle introduced by Heberden—rate of heat loss from a heated body. Only instead of measuring heat loss by the rate of cooling of a previously heated bulb the devices now introduced, the Comf-thermometer of Hill (1923)[8,9] and the Comfortimeter of Phelps (New York State Commission, 1923)[10] both depended on a continuous constant supply of heat to the instrument and a record of the temperature resulting from the balance between this heat input and the loss from the surface of the instrument itself. The Hill device measured the air temperature and the air movement while that of Phelps was covered with a wet cloth so as to be affected by relative humidity as well.

At the same time, the Research Laboratory of the A. S. H. V. E. (1923),[11] introduced an interesting new idea into ventilation practice—that of Effective Temperature. This involved no new instruments but merely the use of a single index expressing the combined effect of temperature, air movement and relative humidity upon comfort, as determined empirically by experiments with human beings.

During the past few years this entire subject has entered upon a new phase with a growing recognition of the importance of the fourth factor governing heat loss from the body—radiation. According to the computations of Aldrich (1928)[12] radiation actually accounts for nearly half the total heat loss from the normally clothed body and yet none of the instruments mentioned above measure this factor with any degree of accuracy. In addition to the crude Radif-thermometer mentioned previously, three interesting instruments have been described to record radiation effects. The simplest of these, which takes account of air temperature and radiation only, is the Globe thermometer of Vernon (1923).[13] This is simply an ordinary mercury thermometer enclosed in a 6-in. or 9-in. sphere of blackened copper. The other two new instruments measure air movement as well, since they are maintained at a temperature above that of the atmosphere.

The Coolometer of Weeks (1931)[14] consists of a copper spool about 1 in. in diameter and 2 in. long, wound with small copper and manganin wires to constitute a combined heating cell and resistance thermometer. A thin shell is fitted over the spool to serve as a cooling

[1]Director, John B. Pierce Laboratory of Hygiene.
[2]Associate Director, John B. Pierce Laboratory of Hygiene.

[3]Lindsay, H. B., (Note) *Heating, Piping and Air Conditioning, 5,* 311, 1933.

[4]Heberden, W., An Account of the Heat of July, 1825; Together with Some Researches Upon Sensible Cold, *Phil. Trans., Roy. Soc. of London, 1,* 69, 1826.

[5]Hill, L., The Physiology of Open-air Treatment, Lancet, *184,* 1283, 1913.

[6]Hill, L., Griffith, O. W., and Flack, M., The Measurement of the Rate of Heat-loss at Body Temperature by Convection, Radiation and Evaporation, *Phil. Trans., Roy. Soc., London,* Series B. *207,* 183, 1916.

[7]Hill, L., The Science of Ventilation and Open-air Treatment, part I, Med. Res. Council, Spec. Report Series 32:46. 1919.

[8]Hill, L., The Science of Ventilation and Open-air Treatment, part II, Med. Res. Council, Spec. Report Series 52, 1923-a.

[9]Hill, L., Kata-thermometer in Studies of Body Heat and Efficiency. Med. Res. Council of Great Britain, Special Report Series 73, 48, 1923-b.

For presentation at the 41st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buffa'o, N. Y., January, 1935.

[10]New York State Commission on Ventilation, E. P. Dutton Co., New York, N. Y., 1923.

[11]Houghten, F. C. and Yaglou, C. P., Determining Lines of Equal Comfort, A. S. H. V. E. TRANSACTIONS, Vol. 29, 1923.

[12]Aldrich, L. B., Study of Body Radiation, *Smithsonian Misc. Coll.,* 81, 82, 1928.

[13]Vernon, H. M., *Proc., Inst. Heating and Vent. Engineers, 31,* 100, 1932.

[14]Weeks, W. S., A New Instrument for Measuring Cooling Power: The Coolometer, *Jour. Ind. Hyg., 13,* 261, 1931.

surface, silvered to measure the influence of air temperature and air movement, or blackened to include radiation effects as well. The instrument is maintained at a constant temperature and the current necessary to produce this result gives the measure of cooling power.

A more elaborate instrument operating on the same general principles had been described in England by Dufton (1930)[15] and more fully by the same author two years later (Dufton, 1932).[16] This device, called the Eupatheoscope, is a blackened copper cylinder 22 in. high and 7½ in. in diameter, preliminary studies having indicated that a body of approximately this size was necessary to imitate the relative effects of radiation upon the human body. The surface of the cylinder is maintained at 75 F, by two electric bulbs controlled by a thermostatic element. A portion of the current supplied to the lamps passes through a coil wound round a mercury thermometer and this thermometer furnishes a measure of the heat input necessary to maintain the skin temperature of the cylinder at the desired point. This thermometer reads on an arbitrary scale of so-called equivalent temperature. A simplified form of this instrument is extensively used in England as a device for controlling radiant-heating systems in actual operation.

The Eupatheoscope is an ingenious and valuable instrument for the measurement of cooling effects in ordinary occupied rooms; but its range is rather narrowly limited. It obviously ceases to operate when the environment is so warm as to raise the temperature of the cylinder above 75 F, or so cool as to exceed the power of the lamps to maintain that temperature of 75 F. Willard, Kratz and Fahnestock (1933)[17] have made certain modifications of the instrument, which render it somewhat more flexible.

For the purposes of the studies planned in the John B. Pierce Laboratory the range of the Eupatheoscope was far too narrow, since it was desired to include somewhat extreme conditions of both hot and cold environments. It was decided, therefore, as suggested by Prof. A. V. Hill (1932), to reverse the Dufton principle of maintaining a constant surface temperature and measuring heat input and, instead, to supply a constant heat input and record the surface temperature resulting under various environmental conditions. The instrument devised along these lines, which has been called the Thermo-Integrator, has proved so useful that it seems desirable to describe it at this time.

Description and Mode of Operation of the Thermo-Integrator

The Thermo-Integrator, Fig. 1, consists of a hollow copper cylinder 24 in. long and 8 in. in diameter with hemispherical ends, the metal being approximately 0.05 in. in thickness. Each end is provided with a cylindrical nipple ½ in. in bore and about 1 in. in length, the nipples being threaded on the outside. Over each end nipple is fitted a brass cap suitably threaded and provided with a central opening. Through one of these brass caps there passes a piece of brass rod about ⅛ in. in diameter while through the other a piece of copper tubing ¼ in. in diameter is inserted. Both the rod and the tube are insulated from the shell by means of rubber stoppers through which they are inserted. A suitable length of No. 24 Nichrome wire serves to join the two inside ends of the brass rod and copper tube. The tube and rod in this way serve as terminals for the Nichrome filament which is the heating source for the instrument. In addition, the cylinder is evacuated to

Fig. 1—The Thermo-Integrator

less than 0.1 per cent of an atmosphere through the copper tube, a procedure which aids in maintaining a rather uniform surface temperature by minimizing convection currents and air stratification within the instrument. The heater circuit is connected to two 6-volt storage batteries and adjusted to dissipate 21.5 w of energy. This is equivalent to a heat dissipation by the instrument of 17.5 Btu per square foot per hour.

The copper cylinder was made in two parts from two large sheets of copper by the well known metal spinning process, after which the parts were trimmed to dimensions. The two halves of the cylinder were then soldered together, after which the whole was electroplated so as to make an air-tight vessel. The two nipple ends previously described were turned on a lathe separately and then soldered in place prior to electroplating.

Eight iron-constantan thermocouples (Fig. 2) were distributed over the periphery of the Thermo-Integrator in order to record its surface temperature. The thermo-

[15]Dufton, A. F., The Effective Temperature of a Warmed Room, *Phil. Mag.* 9, (59), 858, 1930.

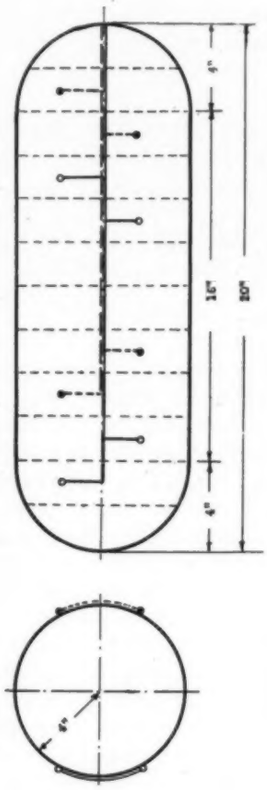
[16]Dufton, A. F., The Temperature of a Warmed Room. The Eupatheoscope, Department of Scientific Research of Great Britain, Building Research. Technical Paper No. 13, 1932.

[17]Willard, A. C., Kratz, A. P. and Fahnestock, M. K., The Application of the Eupatheoscope for Measuring the Performance of Direct Radiation and Convection in Terms of Equivalent Temperature, A. S. H. V. E. Journal Section, *Heating, Piping and Air Conditioning*, 5, 369, 1933.

couples were arranged in a parallel circuit and connected to an automatic recorder so as to obtain the average surface temperature. In this manner it was easily possible to obtain a continuous record of the average surface temperature of the Thermo-Integrator.

The mode of operating the Thermo-Integrator is exceedingly simple; the steps may be enumerated as follows:

(1) Make proper electrical connections to the filament of the Thermo-Integrator to supply a constant energy input. Two 6-volt storage batteries, floated by a 4-amp Tungar battery



Physical Dimensions

Length of Thermo-Integrator	24"
Diameter	8"
Surface Area	603 sq.in. or 4.187 sq. ft.
Mass	5640 grams or 12.43 lbs.
Density (Copper)	8.89 gms/cc.
Specific Heat	0.094

Heating Circuit

Heater Wire	#24 Nichrome
Diameter	0.020"
Length of Heater Wire	60 cms.

Fig. 2—The Thermo-Integrator, showing the location of thermocouple elements

charger were used to supply a constant current of 2.5 amp. The voltage drop across the filament is determined from time to time to insure the proper wattage input to the instrument.

(2) Set the instrument up under the atmospheric conditions the effect of which is to be recorded and allow time (ordinarily about half an hour) for equilibrium to be established.

(3) Determine the surface temperature of the Thermo-Integrator. Make ambient air temperature measurements.

Values and Limitations of the Instrument

The surface temperature of the Thermo-Integrator is obviously affected by the three factors of air temperature, air movement and radiation, and unaffected by the fourth factor, relative humidity. No one has yet had the temerity to devise an instrument which will record all four factors simultaneously although it might be done by adding a moist covering to the Eupatheoscope or the Thermo-Integrator.

If one of the four factors is to be ignored for the present it seems that relative humidity is the one to neglect. Evaporation, while exerting a major influence at high temperatures, accounts within the ordinarily comfortable temperature range for less than a quarter of the total heat loss from the body and rather wide variations in relative humidity within this temperature range cause relatively slight changes in this proportion. At an atmospheric dry-bulb temperature of 70 F, with 20 per cent relative humidity, the evaporative heat loss from the body of an average man is about 115 Btu per hour, while at the same atmospheric temperature with 95 per cent relative humidity, it is reduced only by 30 Btu per hour (Houghten, Teague, Miller and Yant, 1931).[18] Thus, an enormous change in relative humidity has only a moderate effect upon total heat loss.

It is hoped, of course, that the Thermo-Integrator may serve as a reasonably satisfactory instrument for representing the total effect of the environment, excluding relative humidity, upon human health and comfort. Comparative studies of an elaborate nature are now under way to determine this point. It may be noted, however, that several factors suggest the probability that the instrument may be serviceable in this way. In general, its heat input of about 17 Btu per square foot of surface area corresponds closely to the heat production of the human body per square foot of surface area. The actual surface temperature of the Thermo-Integrator is of course far more variable than that of the human body, being 10-12 deg above that of the air when there is no marked air movement and slight radiation effect. It seems probable that the instrument may tend to overemphasize radiation effects (since in still air it loses two-thirds of its heat by radiation, while Aldrich's figures would suggest that the human body loses by radiation only three-fifths of the total heat loss minus the evaporative heat loss). It is also very markedly influenced by air movement. How valuable the Thermo-Integrator may be as a general measure of atmospheric comfort, only subsequent studies will determine.

It is certain, however, that this instrument has very great value from another angle—as a means of recording the total radiation effects of a given environmental condition. In an ordinary room, variations in radiation effects, from windows and walls, from sources of heat and light, and from the bodies of other occupants, are extremely variable and almost impossible to sum up accurately by any method of direct measurement. The Thermo-Integrator automatically summates all such radiation effects and, after calibration, can be used to measure them after subtracting from its total heat loss the losses which would be caused by the temperature and velocity of the air as directly measured under the given conditions. It is largely on account of its value for this purpose that the instrument is described at this time.

[18]Houghten, F. C., Teague, W. W., Miller, W. E. and Yant, W. P., Thermal Exchanges Between the Bodies of Men Working in the Atmospheric Environment, *Amer. Jour. Hyg., 13,* 415, 1931.

Unit Air Cooling Equipment Classified by NRA

The National Recovery Administration has approved the request of the *Unit Air Conditioner Manufacturers' Association* to have all air conditioning machinery and incidental equipment built with or adapted for a cooling or refrigerating medium classified under the Refrigerating Machinery Industry Code.

This action is an important and constructive contribution to the coordination of closely related industries under one code rather than under several separate codes.

A Laboratory Study of Minimum Ventilation Requirements: Ventilation Box Experiments

By W. H. Lehmberg[1], A. D. Brandt[2] and Kenneth Morse[1], *(NON-MEMBERS)*
Boston, Mass.

This paper is the result of research conducted at Harvard University, School of Public Health, in cooperation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

THE purpose of this work was to study by laboratory tests the relationship between the intensity of body odor in an occupied space and the quantity of fresh air supplied, the factors affecting odor intensity, and finally the possibility of utilizing odor intensity as an index of minimum ventilation requirements.

It is generally recognized that in crowded and poorly ventilated rooms the air acquires a disagreeable odor which is especially repulsive upon entering from clean outside air. Just what is the characteristic "human smell" we do not know. It is believed to be a compound odor arising from emanations of sebaceous secretion, especially when personal hygiene is defective, from decaying teeth, infected tonsils, from the stomach, bowels, skin, etc.

There is considerable difference of opinion whether pollution of room air by such offensive odors should be considered a public health matter. Many health authorities are of the opinion that, while such odors may be offensive, they are not directly poisonous nor are they known to cause any specific and lasting disease. The odor, nevertheless, may be so offensive as to cause reflex digestive and other systemic disturbances which may interfere with growth and general activity. According to the studies of the New York School Committee on Ventilation[3] there is a reduction in appetite and a disinclination for physical work.

Fortunately, or unfortunately, the occupants of a smelly room lose their ability to recognize the odor soon after entering, but this does not necessarily mean a loss of effect. Breathing fresh air restores the ability to recognize the odor anew. To obtain a comparison of odor intensities, therefore, one should always pass directly from fresh air to the odorous air and note the odor sensation immediately. For an excellent brief discussion of the anatomy and physiology of the sense of smell, the reader is referred to a recent review by Fair.[4]

Next to the thermal condition of the air as indicated by its temperature and humidity, the intensity of odor observed upon entering an occupied room from outside fresh air furnishes the most valuable evidence of good or bad ventilation. The general agreement is that a certain amount of fresh air from outdoors is necessary at all times in order to dilute the odoriferous matter to a concentration that is not objectionable. The amount needed varies so much under various conditions that it seemed advisable to study the problem first in the laboratory where the various factors can be controlled. The results of the present study are, therefore, limited to the laboratory conditions under which they were obtained.

[1]Graduate Student, Harvard Engrg. School, Dept. of Sanitary Engrg.
[2]Graduate Student, Harvard School of Public Health, Dept. of Indust. Hygiene.
[3]Report of the New York State Commission on Ventilation, E. P. Dutton Co., New York, 1923.
[4]Fair, G. M.: On the Determination of Odors and Tastes in Water, *Jour. of New England Water Works Ass'n*, XLVII, Sept., 1933.
For presentation at the 41st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buffalo, N. Y., January, 1935.

Apparatus

The experiments were carried out in the psychrometric laboratory of the Harvard School of Public Health. Male subjects of various body weights and heights were confined, one at a time, in an airtight wooden box, 1 ft 8 in. × 1 ft 8 in. × 6 ft 4 in. long, which was placed inside the psychrometric chamber and connected to the supply and exhaust system of the air conditioning plant. The interior of the box was painted with three coats of waterproof paint. Two windows on the top admitted sufficient light for reading or writing. Clean outdoor air, conditioned and metered, was introduced at the head end of the box through a 4 in. galvanized duct, and distributed uniformly along the length of the box by the use of a U-shaped perforated manifold inverted over the head of the subject. The used air was exhausted through a similar pipe, at the foot end of the box, leading to the exhaust fan of the psychrometric room.

Suitable holes having the shape of the nose were cut in the exhaust pipe for smelling the air coming out of the box.

The ventilation rate through the box was measured by the use of calibrated orifices in the supply pipe. It was controlled by means of a damper and by regulating the speed of an auxiliary direct-current fan unit.

Dry and wet bulb thermometers in the supply and exhaust pipes, close to the box, showed the increase in temperature and moisture content of the ventilating current in passing through the box. In addition we recorded the temperature of the air at various points inside the box in order to obtain the average box temperature.

Procedure

Trained subjects came to the laboratory, one at a time, about 9 a. m. or 1 p. m. and lay quietly and comfortably on a cot or in the open box for a preliminary resting period of one-half to two hours, depending upon previous activity. At the end of this period the subjects were weighed and closed in the box where they lay against the bare wooden bottom with only a small pillow

under the head. The desired test conditions being fixed in advance, the subjects remained in the box until the odor intensity, temperature, and humidity of the air coming out of the box reached a steady state. This usually took one to two hours depending upon the rate of air flow and temperature. In some experiments the test period was prolonged to four hours or more.

The subjects in the box kept records of their own pulse, body temperature, respiration rate, comfort, and general impression as to air quality. Their spare time was usually taken up by reading. Altogether 10 men, between the ages of 20 and 37 years, served as subjects. They were apparently healthy and had normal personal habits.

Observations of odor intensity were made by 15 trained judges, including smokers and non-smokers, who first smelled the odorless air in the psychrometric room (air circulation through room at least 750 cfm per person all from outdoors) for a few minutes, and then the air coming out of the box, through the openings provided on the side of the exhaust pipe. One to three breaths at the most gave full response to odor. The judges were employes of the school. As many of the 15 as it was possible to get hold of in each experiment were called in when needed. They voted confidentially according to the scale in Table 1, without knowing particulars about the test conditions. Their records were finally averaged for each test and the mean odor intensity plotted against the corresponding air flow.

Table 1—Sensory Intensity Scale of Body Odor

Odor Intensity Index	Characteristic Term	Qualification
0	None	No perceptible odor.
½	Threshold	Very faint, barely detectable by trained judges; usually imperceptible to untrained persons.
1	Definite	Readily detectable by all normal persons but not objectionable.
2	Moderate	Neither pleasant nor disagreeable. Little or no objection. Allowable limit in rooms.
3	Strong	Objectionable. Air regarded with disfavor.
4	Very strong	Forcible, disagreeable.
5	Overpowering	Nauseating.

The zero base of the scale is, of course, not absolute but it represents the sense of smell of the air in the psychrometric room and in the supply to the box, the latter being a portion of the air stream leading to the psychrometric room and differing from it in temperature only. As the entire air conditioning system, including ducts and registers had been washed thoroughly prior to this study, it would be reasonable to assume that the air in the psychrometric room did not differ appreciably in smell from the air out of doors. An odor strength of zero was not recorded in any box experiment with air flows up to 50 cfm, as compared with the controlled air flow of 750 cfm per person in the psychrometric room.

Odor index No. 2 depended to some extent on the sensitivity or tolerance of the judges to smell, but with the exception of two women, the records of the men agreed within ± ½ point or less on the 5 point scale. As a rule the two women judges voted higher than the men. Before training, the judges preferred to record odor strength in whole numbers, but afterwards they felt confident enough to vote to the nearest half division on the scale. Likewise, the threshold intensity index declined from 1 before training to ½ after training.

For full response, the smelling of the odor had to be done in one to three breaths at the most, and the mucous membranes had to be neither too moist nor too dry. Blowing the nose lightly just before smelling helped in all cases.

It was often necessary to wash with soap and rinse out thoroughly the inside surfaces of the box, fan, and exhaust duct, so as to begin all experiments with an odorless box. This is an important point because it has a bearing on the claim sometimes made to the effect that odors in occupied rooms cannot be controlled by ventilation. A properly ventilated room never gets to that stage. On the other hand, a poorly ventilated room would require many hours of flushing with clean air to get rid of the odor absorbed by the walls, furnishings, clothing, etc.

In parallel with these observations a series of physiological tests was carried out, the results of which will be presented in another paper.

Results

Odor Strength in Relation to Ventilation Rate and Body Surface Area

In Fig. 1 are shown the results of all experiments with box temperatures between 65 and 72 F. Proceeding from the point of maximum odor intensity 5, which corresponds to air flows under 1 cfm, the odor intensity decreases rapidly as the ventilation rate increases to about

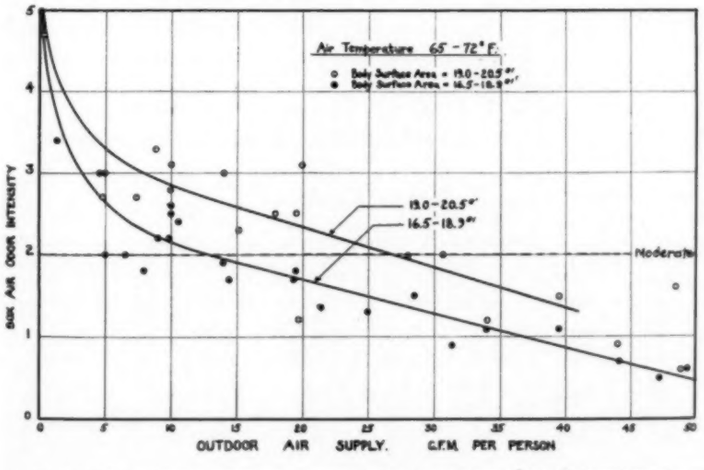


Fig. 1—Odor intensity in relation to ventilation rate and body surface area

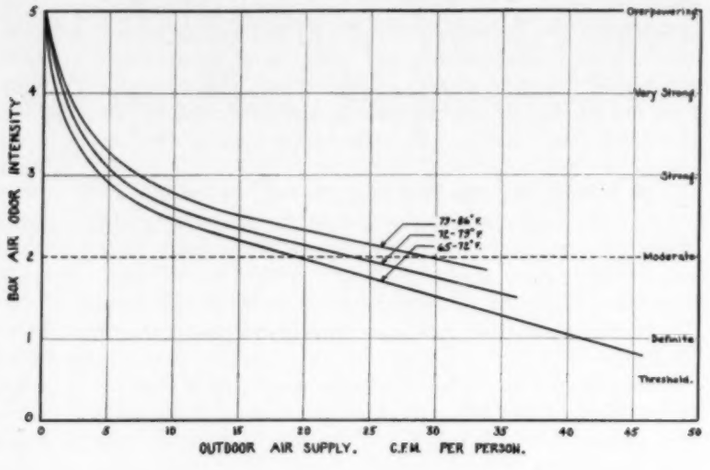


Fig. 2—The influence of air temperature on odor intensity

10 cfm. After that the decrease in odor intensity is gradual, varying almost inversely as the air flow to the first power. According to Fig. 1, the minimum outdoor air supply per person necessary to prevent the accumulation of objectionable body odor in the air of the box varied from 13 cfm for subjects with body surface areas between 16.5 and 18.9 sq ft, to 27 cfm for surface areas between 19.0 and 20.5 sq ft. The average adult has a surface area of 19.5 sq ft.

From this it would seem that body surface area is a very important factor affecting the amount of odoriferous matter given up to the air. The relationship is now being studied more thoroughly in ordinary rooms using school children for subjects as well as adults.

Complete elimination of odoriferous matter in the air of the box was never attained in any of the experiments with ventilation rates up to 50 cfm. This comparatively high air supply was barely sufficient to reduce the odor intensity to the threshold (intensity index ½).

The variability of the results, as shown by the dispersion of individual points from the general trend, is to be expected in tests of this nature. The frame of mind, personal hygiene, type and amount of food eaten, days between last bath and change of underwear to the day of test, proved to be factors influencing the results. Some of these were controlled, more or less, by selecting subjects of normal habits; others by training.

In experiments on a larger scale, such as the ones that are now being carried on in ordinary rooms with as many as 14 subjects at a time, the personal factors seem to be so nicely balanced as to yield very much better results.

The Influence of Temperature on Odor Intensity

In Fig. 2 the results of all experiments are classified in three groups according to the temperature of the air in the box. A rise in temperature is associated with an increase in odor intensity, owing probably to increased perspiration, but the effect is not so great as is generally believed. The minimum ventilation requirements for the three different temperature ranges are about 20 cfm at 65-72 F, 24 cfm at 72-79 F, and 30 cfm at 79-86 F, under the experimental conditions. The last air flow is substantially that demanded by ventilation laws and codes in several states and cities of the U. S. A.

Humidity in Relation to Odor Intensity

Little confirmation could be found for the general belief that high humidities intensify the odor of occupied rooms. As can be seen in Table 2, the difference in odor intensity at relative humidities over 40 per cent (40-90 per cent) and under 40 per cent (15-39 per cent) is too small to deserve mention.

Air Quality in Relation to Air Supply

Despite the high odor intensities reported by the judges, who compared the fresh air going into the box with the used air coming out of the box, the subjects inside the box were as a rule unconscious of odor, owing to olfactory adaptation, except for a sensation of staleness, flatness, or lack of freshness in the air as perceived by the sense of smell. Such sensations constituted, in fact, the chief criterion of bad air in the opinion of the subjects. Complaints of this sort were most frequently associated with ventilation rates under 5 cfm even when the air was neither too warm nor too cool. A few of the subjects claimed to have actually smelled a weak body odor throughout the test period with ventilation rates up to about 10 cfm.

Table 3 shows the impressions of the subjects as to air quality under various ventilation rates. Since air

Table 2—Influence of Humidity on Odor Intensity (Judges' Impressions)

Outdoor Air Supply cfm per Person	Average Box Temperature: 65-72 F — R. H. under 40% — No. of Observations	Odor Intensity	R. H. over 40% — No. of Observations	Odor Intensity	72-79 F — R. H. under 40% — No. of Observations	Odor Intensity	R. H. over 40% — No. of Observations	Odor Intensity	79-86 F — R. H. under 40% — No. of Observations	Odor Intensity	R. H. over 40% — No. of Observations	Odor Intensity
Under 5	2	3.9	3	3.6	2	3.5	7	3.6	2	4.0	4	4.4
5-15	5	2.6	7	2.3	8	2.8	11	2.3	5	2.8	2	3.4
15-25	4	1.8	3	2.4	4	1.9	4	3.0	1	2.4	4	2.5
25-35	2	1.7	5	1.3	2	1.6	3	1.7	4	2.3	3	1.6
All air flows	13	2.4	18	2.3	16	2.5	25	2.7	12	2.8	13	3.0

Table 3—Subjective Impressions of Air Quality in Relation to Air Supply

(*Under comfortable conditions of temperature and humidity*)

Outdoor Air Supply CFM per Person	Number of Subjects Recording: Excellent	Good	Fair	Poor	Bad	Consensus
1-2	..	..	..	..	2	Bad
4-5	..	1	1	4	..	Poor
6-8	..	2	2	..	..	Fair to good
9-11	2	3	5	1	..	Fair to good
12-16	1	3	1	..	..	Good
17-20	5	3	..	..	..	Excellent
21-35	..	4	1	..	..	Good
36-50	1	1	9	..	..	Fair

quality depends so much upon temperature and humidity, all experiments in which the subjects were conscious of heat or cold, excessive moisture, or dryness, have been excluded from Table 3, with but two exceptions. In the two special cases the ventilation rate was under 2 cfm and therefore, the humidity inside the box was necessarily high. The box temperature was kept at a comfortable degree by cooling the psychrometric room, rather than by excessive cooling of the ventilating current. Such a condition of high humidity would exist in any occupied room when the ventilation rate is reduced to 2 cfm per person, or less, regardless of moisture content in the air supplied. The impression of air quality will therefore be affected by the coexisting factors of excessive humidity and odoriferous matter in the air. To some extent this holds true with air flows up to about 5 cfm, unless the moisture content of the ventilating current is kept low, so as to maintain normal humidities in the occupied space, as was done in the experiments shown in Table 3.

Keeping this in mind, it can be seen in Table 3 that with ventilation rates under 5 cfm the air quality was poor to bad; at 10 cfm it was found to be good or fair, and at about 20 cfm it was excellent. Ventilation rates in excess of 10 cfm impaired the quality of air under the experimental conditions, owing to an indescribable irritating sensation produced by non-chilling drafts. The objection was more serious with chilling drafts, but such experiments are not included in Table 3. This is clearly a limitation of the experimental method; the box was too small for the high air flows. Nevertheless, similar complaints often occur in ordinary rooms with ordinary distribution systems, when the circulation rate exceeds 20 cfm.

The judges' impression of air quality depended largely on the intensity of odor which they could smell to the fullest extent. The less the odor, the better the air. But since there was always a certain amount of odor in the box exhaust under all air flows by comparison with fresh outdoor air, the air of the box was never perfect in the opinion of the judges. This is, of course, a theoretical standard not easily realized in practice.

The outstanding point on which both judges and subjects agreed is that the quality of air is impaired when the outdoor air supply falls below 10 cfm per person even under the most favorable conditions of temperature and humidity. This is well in accord with the experience of the New York State Commission on Ventilation[3] and of Eberle and Raiss.[5]

Summary

A definite relationship was found to exist between intensity of body odor in the air of an occupied space and quantity of fresh air supplied, by confining human subjects in an airtight box and smelling the air coming out of the box.

With ventilation rates of 5 cfm per person or less, the odor was as a rule disagreeable and the air was regarded with disfavor by both judges and subjects, although the latter were not conscious of odor except for a sensation of staleness, flatness, or lack of freshness in the air as perceived by the sense of smell.

Under ordinary room temperatures (65-72 F), the minimum outdoor air supply per person necessary to prevent the accumulation of objectionable body odor in the air of the box varied from 13 cfm for subjects with body surface areas between 16.5 and 18.9 sq ft, to 27 cfm for surface areas between 19.0 and 20.5 sq ft. The average adult has a surface area of about 19.5 sq ft. Humidity had little or no effect.

Complete elimination of odoriferous matter in the air of the box could not be obtained with ventilation rates up to 50 cfm per person.

In the light of this preliminary study it would seem that the intensity of body odor in occupied rooms can be taken as a satisfactory index of minimum ventilation requirements. Practical requirements are now being studied in ordinary rooms using school children as well as adults for subjects.

[3]Report of the New York State Commission on Ventilation, E. P. Dutton Co., New York, 1923.

[5]Eberle, C., and Raiss, W.: Die Heizung von Schulgebauen. Beiheft zum Gesundheits-Ingenieur, 1931, No. 29.

Stoker Manufacturers Elect Officers

J. R. Whitehead, manager of the stoker and research divisions, Fairbanks, Morse & Co., Chicago, was elected president of the *Stoker Manufacturers Association* at the annual meeting held in Cleveland, Ohio, on December 10. He succeeds W. H. Rea, Pres., Detroit Stoker Co., Detroit, who has served since the last annual meeting held in June, 1933. R. B. McClave, Pres., McClave-Brooks Co., Scranton, Pa., was elected vice-president, succeeding Mr. Whitehead. Harry H. Kurtz, Iron Fireman Mfg. Co., was re-elected Treasurer and Marc G. Bluth was re-elected Secretary.

Bishop Humidifier Co. Organized

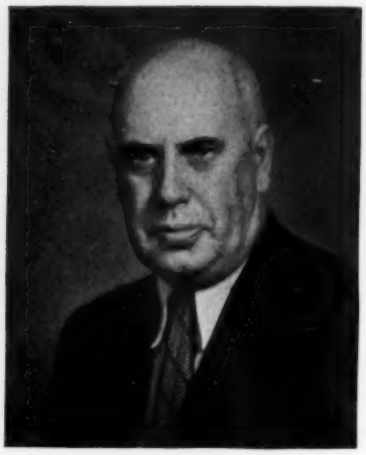
Fred R. Bishop, who has been serving as special representative for manufacturers of different heating equipment, has recently organized the Bishop Humidifier Co., with headquarters at 8011 Dexter Boulevard, Detroit, Mich., and after several months of testing, has placed on the market the Bishop humidifier and the Bishop automatic water pan filler.

Walter Voisinet
Banquet

D. J. Mahoney
Entertainment

B. C. Candee
Finance

C. A. Evans
Transportation



M. C. Beman
General Chairman

Roswell Farnham
Vice-Chairman

P. S. Hedley
Ladies

Mrs. P. S. Hedley
Ladies

L. A. Harding
Reception

J. J. Landers
Registration

C. W. Farrar
Publicity

Buffalo's New City Hall

A. S. H. V. E. Annual Meeting Program

THERE will be many things in Buffalo to attract members of the Society to the 41st Annual Meeting, opening at the Hotel Statler, January 28. The Committee on Arrangements has promised some innovations and under the active leadership of Chairman M. C. Beman, several novel events will be introduced.

All members, ladies, and guests are invited to attend a Get-Together Dinner at 6:30 Monday evening, January 28, after which they will witness a radio broadcast of the Times Variety program from the Terrace Room. This entertainment going on the N. B. C. Red network, presents a full hour of diversified entertainment with a wide appeal. Following the broadcast, dancing will be in order.

At 1:00 o'clock on Tuesday afternoon the ladies will have a luncheon and card party and at the same hour the Past Presidents will attend a special luncheon.

On Tuesday evening, January 29, the Annual Banquet and Dance will be held in the ballroom of Hotel Statler with L. A. Harding as toastmaster. In the brief formal program, presentation of the Past President's emblem will be made to C. V. Haynes and the president elect will be introduced. There will be appropriate music during dinner and plenty of opportunity for dancing.

Members of the Society will start to assemble for the 41st Annual Meeting on January 27, when the Council and various committees will hold meetings in the Hotel Statler. Registration for the meeting will open at 8:30 a. m., January 28, and Pres. C. V. Haynes will call the first session to order at 10 a. m.

Two technical sessions will be held each day and at the conclusion of the meeting on Wednesday afternoon, January 30, the new Council will meet for organization. A fine program of papers has been prepared and there will be ample opportunity for discussion of the subjects presented.

Several papers have resulted from cooperative work with the Committee on Research and the Society's Research Laboratory will report on studies made during the past summer.

The expense of attending the Annual Meeting will be low as hotel rates are moderate and reduced round-trip fares on the railroads have been granted under the identification certificate plan. Room reservation cards and railroad certificates will be mailed to members from the Secretary's office in ample time to make reservations and purchase tickets. The identification certificate is good from all points in the United States and Canada and must be presented to local ticket agents in order to purchase round-trip tickets at a rate of one and one-third fares. One certificate may be used by a Society member and his entire family.

Tickets are good for 30 days and may be used on one route for the going trip and an alternate route returning, if this is specified at the time of purchasing tickets. At the railroad station in Buffalo, members must have tickets stamped before making the return trip. There is a big saving available to members of the Society as their membership gives them the privilege of attending meetings under the reduced fare plan.

At this time the personnel of the Committee on Arrangements for the Western New York Chapter are introduced. Their purpose is to see that everyone has a good time in Buffalo. They want every member to attend the regular functions and enjoy the entertainment program.

Be sure to attend the 41st Annual Meeting in Buffalo, January 28 to 30.

Method of Choosing Location of, Financing and Conducting Meetings of the Society

Resolved: That inasmuch as the Annual and Semi-Annual Meetings of the Society come under the jurisdiction of the Council, the following rules governing the handling of such meetings be adopted by the Council and published in the JOURNAL of the Society at least twice during every year, preferably just prior to each meeting:

1—The Council will select the city in which the Annual or Semi-Annual Meeting is to be held, giving due consideration to the invitations received from Chapters or members as well as to the advisability of so distributing those meetings as to make them of the greatest advantage to the general membership, and to reduce as far as possible the expense of members attending.

2—That an appropriation be made to cover the entertainment or local expenses, incurred in connection with the meeting not exceeding $500.00, the regular meeting expense to be taken care of by the General Fund of the Society in the regular way.

3—That no registration fee or compulsory obligations of any nature be imposed on members or guests.

4—That the purchase of tickets for banquets or for any other form of entertainment that may be provided be entirely voluntary.

5—That the grouping of features and the sale of tickets for group features be discouraged.

6—That the raising of funds from manufacturers of heating apparatus be discouraged.

7—That the display of samples, or of literature, advertising the product of any manufacturer in any way, shape or form, be not permitted at the booths, registration desk, or in or about the meetings.

8—That the distribution of trade papers be entirely at the discretion of the committee in charge.

9—That the local Chaper or local members, be empowered to form a General Committee with such sub-committees as may be required to handle the details of transportation, hotel accommodations, entertainment, finance, etc., and that this General Committee be requested to confer frequently with the Council, through the Secretary of the Society, and to make frequent reports on progress in connection with the various matters being handled by them.

10—That the arrangements of elaborate and costly entertainment features be discouraged.

Adopted at Council Meeting, January 29, 1926.

Hotel Statler,
Buffalo, N. Y.

Program

41st ANNUAL MEETING

AMERICAN SOCIETY OF HEATING & VENTILATING ENGINEERS

Hotel Statler Jan. 28-30, 1935 Buffalo, N. Y.

January 27 (Sunday)

2:30 p. m. Meeting of Council.

January 28 (Monday)

8:30 a. m. Registration.

10:00 a. m. First Session—

Greeting by L. A. Harding
Response by C. V. Haynes, President.
Report of Officers.
Technical paper:
Some Recent Studies on the Ashes in Coal, Prof. T. G. Estep.
Reports of Council Committees.
Report of Committee on Revision of Constitution, By-Laws, and Rules.
Report of Tellers of Election.

2:30 p. m. Second Session—

Report of Committee on Research.
Technical papers:
Cooling Requirements of Rooms in an Unconditioned Office Building, F. C. Houghten and Carl Gutberlet.
Study of Unit Room Coolers in the Research Residence, A. P. Kratz, M. K. Fahnestock, and S. Konzo.

6:30 p. m. Get Together Dinner, Hotel Statler.

7:00 p. m. Meeting Committee on Research.

8:30 p. m. Times Variety Broadcast—Terrace Room. Dancing after Broadcast.

January 29 (Tuesday)

9:30 a. m. Third Session—

Report of Guide Publication Committee.
Technical papers:
Carbon Monoxide Control Instrument for Control of Ventilation in Garages by F. C. Houghten.
The Nature of Ions in Air and Their Possible Physiological Effects, Prof. L. B. Loeb.
Large-Ion and the Small-Ion Content of Air in Occupied Rooms, by G. R. Wait and O. W. Torreson.

1:00 p. m. Past Presidents' Luncheon.

1:00 p. m. Luncheon and Card Party for Ladies.

2:00 p. m. Fourth Session—

Technical papers:
Design of Solar Heaters by Harold L. Alt.
The Thermo-Integrator—A New Instrument for the Observation of Thermal Interchanges, C.-E. A. Winslow and Leonard Greenburg.
A Laboratory Study of Minimum Ventilation Requirements: Ventilation Box Equipment by W. H. Lehmberg, A. D. Brandt and Kenneth Morse.
Report of Ventilation Standards Committee.

7:00 p. m. Banquet and Dance.

January 30 (Wednesday)

9:30 a. m. Fifth Session—

Technical papers:
The Heat Requirements of Buildings, J. H. Walker and G. H. Tuttle.
Tests of Three Heating Systems in an Industrial Type of Building, G. L. Larson, D. W. Nelson, and John James.

2:00 p. m. Sixth Session—

Installation of Officers.
Technical papers:
Study of Summer Cooling in the Research Residence for the Summer of 1934, A. P. Kratz, S. Konzo, M. K. Fahnestock, and E. L. Broderick.
New Business.
Resolutions.

4:30 p. m. Council Meeting.

NOMINATIONS FOR 1935

The Nominating Committee appointed to select candidates for Officers of the Society for the coming year, 1935, submits the following list of nominees:

For President:

JOHN HOWATT, Chicago, Ill.

For First Vice-President:

G. L. LARSON, Madison, Wis.

For Second Vice-President:

D. S. BOYDEN, Boston, Mass.

For Treasurer:

A. J. OFFNER, New York, N. Y.

For Members of the Council:

Three-Year Term

J. F. MCINTIRE, Detroit, Mich.
ALBERT BUENGER, St. Paul, Minn.
F. E. GIESECKE, College Station, Texas
W. E. STARK, Cleveland, Ohio

Respectfully submitted,

NOMINATING COMMITTEE,

W. R. EICHBERG, *Chairman;*
W. G. BOALES, *Acting Secretary.*

In accordance with the provisions of the Amended Constitution, as given, ballots containing the names of the above candidates will soon be sent to the membership for voting upon prior to the Annual Meeting in January.

Art. B-VIII—Section 10. The Nominating Committee shall consist of one (1) member eligible to vote designated by each Chapter, or his alternate also appointed by the Chapter. The Secretary of each Chapter shall certify to the Secretary of the Society on or before January first the names of the member and alternate selected.

The Committee shall meet at the Annual Meeting of the Society at the call of the Secretary of the Society and shall effect its own organization and elect its own Chairman. At the Semi-Annual Meeting of the Society, if possible, the Nominating Committee shall select the nominees for the ensuing year for the offices of President, First Vice-President, Second Vice-President, Treasurer, and four (4) members of the Council. In any event the names of the nominees shall be certified to the Secretary of the Society before September twentieth, with the written consent of each nominee to fill the office for which he has been selected and their names with the offices to which they have been nominated shall be published in the October issue of the JOURNAL.

Art. B-IX—Section 2. The Secretary shall prepare ballots with the names of all candidates and forward them to the members, eligible to vote, at least thirty (30) days before the date of the Annual Meeting.

Nominations for Members of Committee on Research

Although committees of the Society are usually appointed, in view of the great importance of the Committee on Research and the financial responsibility it would be called upon to assume, it has been made more representative of the entire membership of the Society by a process of *election*. The election is governed by the By-Laws for the election of officers, with the single exception that Members of the Committee on Research are nominated by the Council instead of by a Nominating Committee.

In accordance with the Regulations for the Government of the Research Laboratory, adopted at the January, 1919, Meeting, and amended January, 1933, the Council announces the nomination of the following members of the Committee for election to succeed those members whose present terms expire January, 1935:

Three-Year Term

C. A. DUNHAM, Chicago, Ill.
W. L. FLEISHER, New York, N. Y.
ELLIOTT HARRINGTON, Schenectady, N. Y.
A. P. KRATZ, Urbana, Ill.
H. C. MURPHY, Louisville, Ky.

The regulations governing the nomination and election of members of the Committee on Research are as follows:

ARTICLE II—ORGANIZATION

Section 1. *Committee on Research*—There shall be a standing committee known as the Committee on Research, consisting of fifteen (15) members each serving for three (3) years and five (5) retiring each year. The outgoing Chairman if serving in that capacity during the last year of his three (3) year term on the Committee shall without election become an additional member of the Committee on Research for one (1) year.

(a) The Council shall nominate previous to July first of each year five (5) members to fill the vacancies of those retiring at the next Annual Meeting.

(*b*) The nominations made by the Council shall be published in the October issue of the Society's Journal.

(*c*) Any ten (10) members of the Society eligible to vote may present to the Secretary over their signatures, the name of one (1) or more additional nominees for the Committee on Research, provided such name or names are presented at least sixty (60) days prior to the next Annual Meeting, and such additional nominations shall be placed on the ballot opposite the nominations made by the Council.

(*d*) The election shall otherwise conform to the regulations provided for the election of officers of the Society in the Constitution, By-Laws and Rules.

(*e*) Vacancies may be filled by the Council, such persons chosen by the Council to serve until a successor is elected at the next Annual Meeting.

Local Chapter Reports

Cleveland

November 20, 1934. The November meeting was the occasion of a joint gathering with the *Heating, Piping & Air Conditioning Contractors Cleveland Association.* At 7:00 p.m., 167 members and guests of the two groups assembled in the banquet room of the Cleveland Club. During the dinner C. F. Seifried and T. O. Murphy uncovered real harmony at the piano and all joined in the old favorites which were sung.

Walter Klie acted as toastmaster and called the meeting to order at 8:00 p.m., but, before he could introduce the speaker of the evening, the club manager accompanied by "detectives" J. S. O'Gorman and Les Sterns insisted on stopping the proceedings in order to locate a statue which had disappeared from the lobby. The unsuccessful search, however, brought to light some very interesting articles among which were the auditorium patents found on Les Avery and a machine in the clothing of H. M. Nobis which was said to be dangerous.

The prize skit of the evening, A Lesson in Salesmanship, was enacted by Lou Williams and Walter Rhoton. Mr. Williams as Joe Palooka, the star unit heater salesman, ably demonstrated that even though he may have been slightly mixed up on his Btu's, cfm's, and rpm's, his steady gas attack finally left Mr. Rhoton in such a weakened condition that he gave him his shop as well as the order.

Monroe Marks, of the *Heating, Piping & Air Conditioning Contractors Cleveland Association,* took the leading part in a humorous but instructive sketch on the preparation and filing of bids under the new heating code.

Toastmaster Klie finally restored order and introduced A. R. Herske as guest speaker of the evening. Mr. Herske's subject was Prospects for the Heating and Air Conditioning Industry and the speaker's knowledge of the present condition of the heating industry in all sections of the country was clearly indicated.

Mr. Herske gave a timely message in his report of results being obtained under the modernization section, Title I, of the F.H.A. The speaker advised the members that Title II, which provides for new construction should be in effect by January 1. The opportunities which will be afforded the building industry under this act will be reflected in the heating and air conditioning industry in particular.

A rising vote of thanks was extended to the speaker and the meeting adjourned at 10:30 p.m.

October 25, 1934. The meeting was called to order by Pres. M. F. Rather in the Meeting Room of the *Cleveland Engineering Society,* with 24 members and guests present.

Attention was called to the Cleveland Chapter Membership Roster, recently published and distributed to members, and to the various committees appointed. F. A. Kitchen, chairman of the Membership Committee, requested that all members suggest the name of at least one prospective member.

The meeting was turned over to W. E. Stark, who introduced C. V. Haynes, president of the A.S.H.V.E.

President Haynes gave a general report as to the present condition of the Society in regard to membership and finances. He also stressed the improvements to be found in THE 1935 GUIDE and told of the program being prepared for the 41st Annual Meeting of the Society to be held in Buffalo in January.

W. E. Heavey, engineer of the Great Lakes Division, U.S.A. Engineers Corps was presented and gave a very interesting and instructive talk on the amount of work continually being carried on in order to maintain necessary water levels in the entire Great Lakes area. His talk was illustrated with slides, showing some of the harbor improvements in the Great Lakes region and flood control work now in progress in the Central States.

Following these speakers, W. E. Stark entertained with two reels of movies of camp life at the National Guard Encampment at Camp Perry.

The meeting adjourned at 10:25 p.m.

Illinois

November 12, 1934. The November meeting of the Illinois Chapter was held at the Hotel Sherman, with an attendance of 106.

The newly elected president, R. E. Hattis, called the meeting to order at 7:45 p. m. after dinner in the Old Town Room. The minutes of the October meeting were read and approved. Letters from A. V. Hutchinson regarding the Council's plan for deferred payment of dues for those in arrears and announcing the rate of dues for the coming year were presented.

A report was made by the secretary covering the Illinois Chapter's new Annual Limited Chapter Membership.

President Hattis then called upon John Howatt, First Vice-President of the A.S.H.V.E. to introduce the speaker of the evening. Mr. Howatt briefly outlined the field of air conditioning for the next few years, declaring that the industry must look to the remodeling of existing buildings for its business. Owners no doubt will be forced to air condition their properties, in order to hold their tenants. He spoke of the many early experiments in air conditioning conducted by Dr. E. Vernon Hill, the speaker of the evening.

Dr. Hill said he would try to live up to Mr. Howatt's introduction, and then outlined some of the views that he has held for 20 years. He praised the work of J. J. Aeberly, O. W. Armspach, and others in the Chicago Department of Health for their work in 1917 on the synthetic air chart, and its adoption in 1920. Since 1922, he said he has insisted that the term, *fresh air* should not be used when *outdoor air* is meant, it being a question of air quality, and not quantity.

The remainder of Dr. Hill's talk was illustrated by a number of interesting slides, which began with an outline history of early man's contact with the world and his struggle for cleanliness, food and drink, and a healthful air supply. He told of the conquering of plagues by the proper methods of sanitation and care in handling food and water, but stated that sanitary service has made little progress in air treatment until the last few years.

Dr. Hill insisted that outside air is not *fresh,* but contains dust, carbon monoxide, and bacteria, and is not only a nuisance, but a menace to health as well. He furnished numerous health department charts showing the seasonal death rate of pneumonia, heart disease, and nephritis, and stated that these rates are advancing. It is interesting to note that the inverted death curve closely follows the temperature curve throughout the year.

Dr. Hill continued with detailed descriptions of various laboratory tests, covering skin temperatures, personal efficiencies under various temperatures, etc. He concluded that in a warm room, the metabolism rate is low and suggested that our sleeping-rooms be warm. The comfort curve parallels the wet-bulb temperature, he said, and our present comfort charts should be revised. He was also of the opinion that air and oxygen therapy should be considered in the treating of diseases.

Dr. Hill closed his interesting talk by stating again that the fresh outdoor air theory has delayed the development of true air conditioning, because all outdoor air is bad unless it is treated.

An enthusiastic vote of thanks to Dr. Hill closed the meeting at 9:30 p. m.

Massachusetts

November 19, 1934. At the November dinner and meeting at the Engineers Club, Boston, 44 members and guests were present when Pres. R. S. Franklin welcomed C. V. Haynes, Philadelphia, Pa., president of the A.S.H.V.E., who was the guest speaker of the evening.

In his talk President Haynes spoke on Society affairs and discussed its new membership, research activities, financial condition, The Guide, the Annual Meeting to be held in Buffalo, and other matters.

W. T. Jones, past president, spoke on Society matters and D. S. Boyden, treasurer of the A.S.H.V.E., read a report concerning its finances and membership.

At 10:00 p. m. the meeting adjourned, according to the report of W. A. McPherson, secretary.

Michigan

November 12, 1934. Approximately 70 members and guests attended the second meeting of the 1934-35 season of the Michigan Chapter at the Wardell Hotel, Detroit.

Pres. G. D. Winans called the meeting to order and dispensed with the roll call and reading of the minutes.

Treas. R. K. Milward reported $91.00 in the treasury with all current bills paid.

The special speaker of the evening was the Rev. Frederick Siedenburg, executive dean of the University of Detroit and director of the Detroit Regional Labor Board. His topic was Some Aspects of the New Deal.

In discussing the subject, Reverend Siedenburg pointed out that in his opinion the industries of the country would eventually have more government control in the form of restrictions on minimum hours and wages, old age pensions, and unemployment insurance.

Following this talk, J. L. Fuller, chairman of the Field Survey of the Detroit Better Housing Program, briefly outlined the work of the committee in Detroit.

The second part of the meeting was devoted to a technical discussion of stokers. R. L. Beers was the principal speaker and Sabin Crocker and E. E. Dubry took part in the discussion.

Secy. W. F. Arnoldy reports that the meeting adjourned at 10:45 p. m.

Ontario

December 3, 1934. Sixty-one members and guests were present at the regular meeting of the Ontario Chapter at the Royal York Hotel, when dinner was served promptly at 6:30 p. m.

A talk on Oil Burner Units was given by E. T. Whittall, and, following this address, C. W. Farrar, representing the Western New York Chapter, extended a special invitation to the Ontario Chapter to attend the Annual Meeting of the Society in Buffalo, January 28-30. Pres. W. R. Blackhall announced that H. B. Jenney was making arrangements for members to attend this meeting.

President Blackhall then called upon E. H. Gurney to introduce the principal speaker of the evening, Prof. E. A. Allcut, University of Toronto, who spoke on The Design and Development of Concealed Heaters.

Following Professor Allcut's address, there was a splendid discussion covering the various details of the presentation.

Secy. H. R. Roth reports that after the Chapter's appreciation had been expressed to Professor Allcut the meeting adjourned.

November 5, 1934. The regular November meeting of the Ontario Chapter was held at the Royal York Hotel and at 6:30 p. m. dinner was served with 36 members and guests present.

Pres. W. R. Blackhall called upon E. H. Gurney to report on the October meeting of the Council held in Buffalo, N. Y. Mr. Gurney urged as many members as possible to attend the Annual Meeting of the Society in January at Buffalo.

The principal speaker of the evening was J. Keenan, whose subject was It Pays To Be Modern.

M. F. Thomas expressed the Chapter's appreciation of Mr. Keenan's interesting address.

The appointments for the Nominating Committee were M. F. Thomas, representative, and H. H. Angus, alternate, according to the report of the secretary, H. R. Roth.

Philadelphia

November 8, 1934. Pres. W. P. Culbert opened the meeting at the Engineers Club at 8:00 p. m. Secy. W. R. Eichberg was back on the job and read the minutes of the October 11 meeting, which had been prepared for him by M. F. Blankin.

W. B. Hutt, representing the 1935 Welfare Federation Campaign, gave a short interesting talk requesting assistance for this work.

J. D. Cassell reported on the Council meeting held in Buffalo and mentioned particularly the plan for deferred payment of dues in arrears.

A letter from the Nominating Committee was read by President Culbert giving the following list of names to be voted on at the annual meeting of the Chapter in January: J. H. Hucker, *president;* W. R. Eichberg, *vice-president;* H. H. Erickson, *secretary;* and E. N. Sanbern, *treasurer.* In presenting the nominations, the Committee recommended that the office of secretary and treasurer be combined with a corresponding secretary and bookkeeper to be obtained for service from year to year, taking care of all correspondence and financial records of the Chapter.

Following the reading of the Nominating Committee's letter, a motion was presented by A. H. MacDade, asking that a committee be appointed by the President to formulate a resolution covering the suggestion of the Nominating Committee and such changes in the Constitution as would be required to put them into effect, as well as any other changes deemed advisable.

President Culbert took immediate action and appointed the following members to the Committee: F. D. Mensing, *Chairman;* L. P. Hynes, W. F. Smith, M. F. Blankin, and W. R. Eichberg.

The meeting was then turned over to A. C. Caldwell of the Meetings Committee, who introduced Benjamin Adams. Mr. Adams gave a description of a new ventilating fan.

The next paper was presented by L. H. Plum and was entitled Some New Types of Controls. This discussion was very instructive and was well received by the membership.

Something New in the Vacuum Pump was presented by A. D. Harvey, who, in addition to preparing a worthwhile discussion, presented his paper in an able manner.

The meeting adjourned at approximately 10:15 p. m.

St. Louis

December 5, 1934. After dinner, the meeting was called to order by Pres. Paul Sodemann with 27 members and guests present at the Laclede Gas Building.

The minutes of the previous meeting were read and approved and Treas. G. W. Myers gave the yearly report on Chapter finances.

The secretary, A. L. Walters, submitted the report of the Nominating Committee, after which the Chair called for additional nominations from the floor. There being none, it was moved by R. M. Rosebrough, seconded by Mr. Myers, that the Secretary cast a unanimous ballot for the committee nominees, This motion was carried and the officers were elected as listed in the report of the November Chapter meeting.

Mr. Sodemann reviewed activities during his term of office including the Open Forum on Air Conditioning at the Ranken School. He then turned over the gavel to J. W. Cooper, newly elected president, who asked for the cooperation of his fellow members in securing new members and in giving program suggestions. The other new officers were presented and made brief remarks accepting their election.

The secretary read a letter from the Ranken School regarding a second Open Forum on Air Conditioning. After discussion, it was moved by E. E. Carlson and seconded by Mr. Sodemann that the Board of Governors investigate this matter and that the Chapter, subject to the Board's approval, endorse this activity and offer its assistance.

A communication from headquarters office was read regarding the appointment of Nominating Committee representatives and the plan of the Council for deferred payment of dues in arrears.

The following members were appointed to the Auditing Committee: C. E. Hartwein, *Chairman*, L. W. Moon, and E. A. White.

President Cooper introduced D. J. Fagin, who spoke on Gas Heating and illustrated his talk with slides showing many interesting applications.

After a rising vote of thanks to Mr. Fagin, the meeting adjourned.

November 8, 1934. After dinner the meeting of the St. Louis Chapter at the Chase Hotel was called to order by Pres. Paul Sodemann with 43 members and guests in attendance. Included by special invitation were the Board of Directors and Past Presidents of the Engineers Club. Pres. A. C. Willard of the University of Illinois was the guest of honor.

The minutes of the October meeting were read and approved and J. M. Foster, chairman of the Nominating Committee, presented its report as follows:

For President—J. W. Cooper
For First Vice-President—C. R. Davis
For Second Vice-President—G. W. F. Myers
For Treasurer—E. E. Carlson
For Secretary—A. L. Walters
For Board of Governors—Paul Sodemann, C. E. Hartwein, D. J. Fagin, E. A. White

President Sodemann welcomed the guests and introduced President Willard, who spoke briefly but interestingly. The meeting then adjourned to the Engineers Club, where President Willard addressed the joint meeting of the members of this group and the Chapter on Air Conditioning for Human Comfort.

The attendance taxed the capacity of the auditorium and Secretary Walters reports that those who were present enjoyed the very interesting and instructive discourse of the subject.

Southern California

November 13, 1934. One of the most interesting meetings of the Southern California Chapter was attended by approximately 40 members and guests, when the speaker at the luncheon gathering was R. E. Phillips, consulting engineer.

Mr. Phillips has designed some of the outstanding air conditioning installations in the Southern California district and he gave a very complete and interesting discussion regarding the problems encountered by contractors working in the Los Angeles Aqueduct which runs 250 miles through a desert. Mr. Phillips particularly emphasized the devices used to overcome the difficulties of tunnel ventilation and the Chapter looks forward to his return as guest speaker in the near future.

At this meeting the following officers were elected:

President—W. E. Barnum
Vice-President—Leo Hungerford
Secretary—P. C. Scofield
Treasurer—A. G. Orear
Board of Governors—W. H. C. Ness, E. H. Kendall, H. L. Warren

Wisconsin

November 26, 1934. The November meeting of the Wisconsin Chapter was held at the Hotel Pfister, Milwaukee, at 6:30 p. m. After dinner G. E. Hochstein gave the Secretary's report and an informal discussion followed.

The meeting was a very good one and brought the largest attendance of the year, which was largely due to the fact that Prof. G. L. Larson, Madison, Wis., was the speaker.

A paper was given on the Results of Research on Various Methods of Heating an Industrial Type of Building and was illustrated by slides.

The subject was an interesting one and, after the talk, Professor Larson opened the meeting to informal discussion. Because of the excellence of the paper and the ground it covered, many good topics were brought out.

E. A. Jones was appointed representative on the Society's Nominating Committee and Pres. Ernest Szekely will serve as alternate.

A very pleasant meeting adjourned at 10:15 p. m.

October 23, 1934. The October dinner and meeting of the Wisconsin Chapter was held at the Hotel Pfister, Milwaukee.

The minutes were read by the Secretary, G. E. Hochstein, who also gave his report. Prof. G. L. Larson, Second Vice-President, then outlined national activities of the Society.

The next matter of business was an informal report by C. H. Randolph on the Chapter's co-operative research work with Marquette University.

The main subject for the evening was the State Heating and Ventilating Code. Pres. Ernest Szekely opened the discussion and, after an introductory word, turned the subject over to J. G. Shodron, who is the appointed chairman of the State Heating and Ventilating Code Committee. Mr. Shodron in turn asked E. A. Jones, J. S. Jung, C. W. Miller, and Professor Larson to bring up points for and against the present code. After the informal talks, Mr. Shodron made the suggestion that he call on the Industrial Commission in Madison when next he is in that city. This was approved by the Chapter.

The Program Committee approved a November meeting in the latter part of the month, with Prof. G. L. Larson as the speaker.

The meeting adjourned at 9:30 p. m.

September 24, 1934. The first fall meeting of the Wisconsin Chapter was held at 6:30 p. m., at the Pfister Hotel. Dinner was served at 6:30 p. m. and the meeting proper began at 8:00 p. m.

Letters from the headquarter's office were read explaining the Council's plan for deferred payment of dues and announcing the dues rate for 1935. The next matter of business was the Treasurer's report.

As is customary in the Wisconsin Chapter, the first meeting was devoted to tabulate Chapter election returns and announce the results. While various reports were being made, the tellers were counting the ballots and made the following report:

President—Ernest Szekely
First Vice-President—C. H. Randolph
Second Vice-President—A. M. Wagner
Secretary—G. E. Hochstein
Treasurer—Martin E. Erickson
Board of Governors—G. L. Larson, J. S. Jung, V. A. Berghoefer

Following the election of officers, Mr. Szekely, the new Chapter president, delivered a very interesting formal talk on Silica Gel. A great deal of his present research work has been on this subject, and a wealth of information was given.

The meeting adjourned at 9:40 p. m., according to Secretary Hochstein's report.

Western New York

December 17, 1934. The December meeting of the Western New York Chapter was the occasion of the annual bowling and Christmas party, held at the University Club, and presided over by Vice-Pres. W. E. Voisinet.

M. C. Beman, general chairman of the Committee on Arrangements for the Annual Meeting, together with D. J. Mahoney and W. E. Voisinet, committee chairmen, outlined in detail some of the plans for the meeting.

A motion was made by Pres. J. J. Yager, and seconded by C. A. Evans that a committee be appointed to consider a change in the By-Laws concerning the election of officers in May and their installation in June.

Mr. Beman presented the choice of the Nominating Committee for officers for the new year:

President—W. E. Voisinet
First Vice-President—P. S. Hedley
Second Vice-President—B. C. Candee
Secretary—J. J. Landers
Treasurer—R. T. Thornton

Election was postponed until a later meeting.

A very gratifying report of the treasurer was read by B. C. Candee, showing a substantial increase in funds over last year.

November 12, 1934. P. S. Hedley, secretary of the Western New York Chapter, reports that the November meeting was held at the Touraine Hotel with 25 members and guests present for dinner and 32 at the meeting.

M. C. Beman, general chairman of the Committee on Arrangements for the 41st Annual Meeting of the Society to be held in January, gave some very interesting and definite information regarding the program.

J. J. Nesbitt, Jr., Philadelphia, Pa., gave a very fine lecture on A Thermal System of Ventilation Control, which resulted in considerable discussion.

At the request of Joseph Davis, a treasurer's report was received from B. C. Candee, which indicated an improvement in the financial condition of the Chapter.

The meeting came to a close with a rising vote of thanks for the speaker.

Pittsburgh

The annual report of the Pittsburgh Chapter has been prepared and the first meeting of the 1933-34 season was held on October 16, 1933, in the auditorium of the U. S. Bureau of Mines Building. When Pres. G. S. McEllroy called the meeting to order, 35 members and guests were present.

F. A. Gunther presented the speaker, George Fisher, Pennsylvania State Archeologist, who spoke on the Mound Builders of Western Pennsylvania. This indicated that an occasional talk on a non-technical subject can be well received.

The next meeting was held on November 13, 1933, in the U. S. Bureau of Mines Building and 44 members and guests attended. President McEllroy introduced C. T. Ingham, Pittsburgh, who described What an Architect Thinks of a Salesman.

On December 11, 1933, an interesting meeting was held at the offices of F. C. McIntosh. The following officers were elected for the ensuing year:

President—P. A. Edwards
Vice-President—L. B. Pittock
Secretary—J. L. Blackshaw
Treasurer—H. A. Beighel

Twenty-five members were present and a delicious home-cooked dinner was served before the meeting began at 8:00 p. m. Instead of the usual formal talk, a round-table discussion was enjoyed.

On January 5, 1934, the meeting was again held at the U. S. Bureau of Mines and 55 members and guests were present. Standing Committees were appointed and the International Heating and Ventilating Exposition and the 40th Annual Meeting of the Society were discussed.

Pres. P. A. Edwards presented J. F. Lamb, who described many phases and angles of air conditioning, as well as the design of unit air conditioners, in his talk on Providing Comfort with Unit Air Conditioners.

After the usual business and committee reports at the February 19 meeting, members were entertained by M. C. Morgan, W. B. Simpson, and F. E. Eckert, who showed a six-reel motion picture of the progressive steps in the making of wrought iron pipe. They also presented a paper which summarized corrosion and piping difficulties in heating systems and suggested remedies.

At the March 12 meeting, considerable time was devoted to discussing the efforts of the Pittsburgh Chapter of the *A.I.A.* regarding the employment and remuneration of professional and technical men on CWA and other government undertakings.

E. H. Riesemeyer, Jr., introduced the speaker, W. E. Stark, Cleveland, Ohio, who gave a paper on Winter Air Conditioning for the Residence. Mr. Stark traced the evolution of winter air conditioning and progress in the heating field. He ably described the requirements of present winter heating and outlined what he considered an ideal winter conditioning plant.

The next meeting was held on April 9 in the auditorium of the U. S. Bureau of Mines when 50 members and guests were present. Chief among the discussions were the Engineers' Code and the attempt to bring about closer relations between the various engineering groups in the Pittsburgh district.

C. M. Humphreys presented C. A. Reed, combustion engineer, who spoke on the importance of correct coal preparation with related methods of combustion and presented a motion picture showing actual mining operations by modern methods.

The last meeting of the season was held on April 30 and, following the customary routine business and a discussion of the Society's Semi-Annual Meeting at Buck Hill Falls, Pa., F. C. McIntosh gave a brief history of the A.S.H.V.E. F. C. Houghten spoke on the research work in progress at the Society's Research Laboratory at the U. S. Bureau of Mines.

President Edwards introduced C. V. Haynes, president of the A.S.H.V.E. In his spirited and inspiring talk, Mr. Haynes discussed many of the matters which come up for action by a Society president. A complete picture of the business and technical organization of the A.S.H.V.E. and a summary of its financial set-up was given. Mr. Haynes pointed out the many advantages to be gained through membership in an ethical socity such as the A.S.H.V.E., because of its activities, publications, research work, meetings and friendships.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 19 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by January 15, 1935, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	*REFERENCES*	
	Proposers	*Seconders*
ALFAGEME, BRAULIO, Owner-Manager, Air Cond., Braulio Alfageme, Ing., Madrid, Spain.	H. B. Howe *(Non-Member)* Nicasio Guisasola *(Non-Member)*	Jesus de Lasala *(I.C.A.I.)* Enrique Gutierrez *(I.C.A.I.)*
BICHOWSKY, F. R., Engr., Surface Combustion Corp., Toledo, O.	H. B. Hull W. L. Fleisher	John Everetts, Jr.
BORUCH, E. R., Sales Engr., Dallas Power & Light Co., Dallas, Texas.	G. H. Meffert C. N. Witmer	W. H. Moler M. L. Brown *(ASRE)*
BOYKER, R. O., Partner, Mac Boyker & Son, Kent, Wash.	W. W. Cox S. D. Peterson	K. F. MacLeod William Mallis
CARY, E. B., Vice-Pres., John Paul Jones, Cary & Miller, Inc., Cleveland, O.	Philip Cohen M. F. Rather	F. A. Kitchen
GERARD, G. G., Student, Pratt Institute, Brooklyn, N. Y.	J. W. Hunter *(Non-Member)* N. W. MacKenzie *(Non-Member)*	Joseph Phyl *(Non-Member)* Thomas Militello *(Non-Member)*
GERMAIN, OSCAR, Foreman, Germain Frère Ltd., Three Rivers, Quebec, Can.	Jules Caron *(Non-Member)* Ernest Denoncourt *(Non-Member)*	Z'eph Lambert *(Non-Member)* J. A. Boisjoli *(Non-Member)*
HERRICK, LEO, Mgr., Crane Co., Ft. Smith, Ark.	W. A. Russell D. H. Painter *(Non-Member)*	C. V. Haynes F. H. Gaylord
KICZALES, M. D., Assoc. Mech. Engr., U. S. Army Motion Picture Service, Washington, D. C. LOCKHART, H. A., Engr., Bell & Gossett Co., Chicago, Ill.	G. S. Frankel A. E. Beitzell, Jr. E. J. Gossett R. E. Moore	W. A. Danielson *(Non-Member)* F. Simpson *(Non-Member)* F. B. Rowley A. B. Algren
MOORE, H. W., Air Cond. Engr., Frigidaire Corp., Dayton, Ohio.	Herman Worsham J. J. LaSalvia	C. E. Lewis A. H. Kirkpatrick
MUSGRAVE, M. N., Pres., Harrison Sales Co., Seattle, Wash.	S. D. Peterson D. C. Griffin	Lincoln Bouillon M. J. Hauan
PERKINS, R. C., Sales Engr., Ilg Elec. Vtg. Co., Memphis, Tenn.	E. J. D'Imor R. H. Hoshall	A. G. Sutcliffe M. K. Arenberg *(Non-Member)*
RIVARD, M. M., Mfrs. Rep., Rivard Sales Co., Kansas City, Mo.	F. F. Dodds E. K. Campbell	W. E. Gillham Carl Clegg
SCHURMAN, J. A., Air Cond. Engr., York Ice Machinery Corp., Cleveland, Ohio. SHARP, H. C., Mgr., Smith Oil & Refining Co., Rockford, Ill.	H. B. Matzen F. R. Dickenson G. E. Merwin J. H. Bailey	K. W. Schick Philip Cohen D. J. Stewart J. T. Atwood *(Non-Member)*
SMITH, W. D., Bryant Heater Co., Cleveland, Ohio.	J. E. Kinner W. E. Stark	F. A. Rodgers
WALLACE, J. B., Dist. Repr., Taco Heaters, Inc., Detroit, Mich.	R. K. Milward H. E. Paetz	A. P. Darlington E. L. Hogan
WARD, F. J., Htg. & Piping Contractor, The F. J. Ward Co., Cold Spring, Ky.	W. C. Pistler H. A. Pillen	C. E. Hust E. B. Royer

Candidates Elected

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and ballotted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

ARCHER, D. M., Dist. Sales Repr., Sarco Co., Inc., Boston, Mass.

AVERY, L. T., Pres., The Avery Engrg. Co., Cleveland, Ohio.

CARTER, Doctor, Htg. & Sanitary Engr., Shanghai, China.

DAVISON, R. L., Director of Research, John B. Pierce Foundation, New York, N. Y.

DEWEY, R. P., Chief Engr., Temp. Control, Barber-Colman Co., Rockford, Ill.

FITZGERALD, M. J., Treas. & Gen. Mgr., Standard Asbestos Mfg. Co., Chicago, Ill.

LEDNUM, J. M., Chief Engr., Silica Gel Corp., Baltimore, Md.

MACCUBBIN, H. A., Htg. Contractor, H. A. Maccubbin, Inc., Baltimore, Md.

MANAHAN, J. E., Air Cond. Engr., Kelvinator Div., Witte Hardware Co., St. Louis, Mo.

MAWBY, PENSYL, Service Engr., Lehigh Navigation Coal Co., New York, N. Y.

MCQUAID, D. J., Consulting Engr., Denver, Colo.

PABST, C. S., Pres. & Mgr., Adams Engrg. Co., Inc., New York, N. Y.

STRAUCH, P. C., Sales Engr., The Henry Furnace & Foundry Co., Edgewood, Pittsburgh, Pa.

WETZELL, H. E., Chief Engr., The Smith & Oby Co., Cleveland, Ohio.

WOODMAN, L. E., Pres., Woodman Appliance & Engrg. Corp., Jefferson City, Mo.

ASSOCIATE

DISNEY, M. A., Co-Partner, Disney-Leffel Co., Kansas City, Mo.

JUNIOR

ALLAN, N. J., Asst. to Mgr., Kansas City Pump Co., Kansas City, Mo.

EPPLE, A. B., Student Engr., B. F. Sturtevant Co., Hyde Park, Mass.

FREY, G. O., Engr. (Asst. Dept. Head), Warner Bros. Theaters, Inc., New York, N. Y.

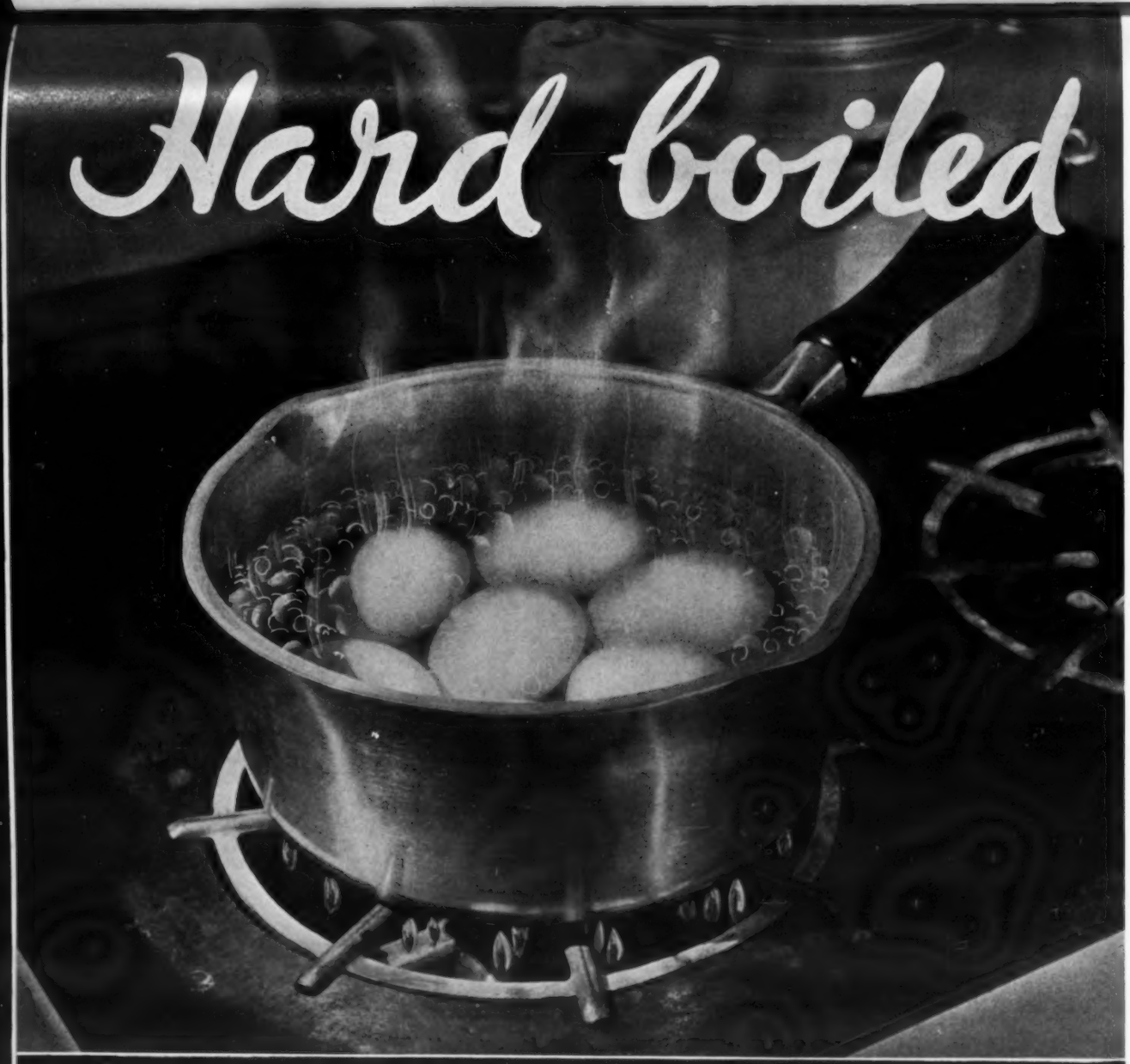
GREER, W. R., Air Cond. & Ill. Engr., Ark. Power & Light Co., Pine Bluff, Ark.

HATEAU, W. M., Draftsman, Sherron Metallic Corp., Brooklyn, N. Y.

SHELDON, W. D., Jr., Chief Engr., Sheldons Ltd., Galt, Ont., Can.

STUDENT

RILEY, R. C., Student, Pratt Institute, Brooklyn, N. Y.



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Handsome sturdy cabinets finished in dark brown with morocco texture, high efficiency streamline tubes, patented fin design, quieter operation, full-floating element mounting, elimination of intertube expansion stresses are just a few advanced engineering features of Fedders Series 3 Unit Heaters. Catalog 527 takes them apart for you, write for your copy.

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Revere Metals are as *Modern* as Air Conditioning

Revere Copper Water Tube

Revere Copper Water Tube is recommended for heating lines, refrigerant lines, water lines, heat control lines and for other heating and air conditioning piping. This tube is seamless, 99.9% pure copper, completely deoxidized and possesses a gun-barrel finish inside, reducing friction losses to the minimum and as a result saving fuel on heating lines and current and wear on the pump on fluid lines.

It is furnished in three types known as "K", "L", and "M" which meet Government and A. S. T. M. specifications for this product. Dimensions and weights are given in the table below.

In general the Type "K" is used where corrosive conditions are severe, such as in underground service and the types "L" and "M" wherever corrosive conditions are normal. For most heating work, Types "L" or "M" will be found satisfactory.

Types "K" and "L" are furnished in both hard and soft tempers. The hard temper is used for new and exposed work and the soft temper for hidden replacement work, and for those uses where flexibility is essential. Type "M" is furnished in hard temper only.

Fittings—This tube is joined with Streamline soldered fittings or with any standard make of compression fittings.

The Streamline fitting is so made that the tube fits against a shoulder on the inside of the fitting, the height of which is approximately the same as the wall thickness of the tube. This produces uninterrupted flow, reduces friction to an absolute minimum and prevents partial stoppage due to the lodgment of sediment, etc.

Revere Copper Water Tube cannot rust, so that it is not necessary to install oversize pipe. Life-time service is assured for all uses, including steam return lines where there is considerable corrosive action. The flow is smooth and uninterrupted and turbulency and noises are reduced to a minimum.

In general, pipe covering can be one size smaller with Copper Water Tube than for the same size S. P. S. pipe because of the smaller outside diameter of the tube.

STANDARD DIMENSIONS AND WEIGHTS

Size in In.	O. D. in In.	TYPE K Underground Services and General Plumbing Purposes Wall Thickness In.	TYPE K Wt Lb per Ft	TYPE L For General Plumbing Purposes Wall Thickness In.	TYPE L Wt Lb per Ft	TYPE M (*Hard Temper*) For General Plumbing Purposes with Soldered Fittings Only Wall Thickness In.	TYPE M Wt Lb per Ft
⅛	.250	.032	.085	.025	.068	.025	.068
¼	.375	.032	.133	.030	.126	.025	.106
⅜	.500	.049	.269	.035	.198	.025	.144
½	.625	.049	.344	.040	.284	.028	.203
⅝	.750	.049	.418	.042	.362	...	...
¾	.875	.065	.641	.045	.454	.032	.328
1	1.125	.065	.839	.050	.653	.035	.464
1¼	1.375	.065	1.040	.055	.882	.042	.681
1½	1.625	.072	1.360	.060	1.140	.049	.940
2	2.125	.083	2.060	.070	1.750	.058	1.460
2½	2.625	.095	2.920	.080	2.480	.065	2.030
3	3.125	.109	4.000	.090	3.330	.072	2.680
3½	3.625	.120	5.120	.100	4.290	.083	3.580
4	4.125	.134	6.510	.110	5.380	.095	4.660
5	5.125	.160	9.670	.125	7.610	.109	6.660
6	6.125	.192	13.870	.140	10.200	.122	8.910
8	8.125	.271	25.900	.200	19.290	.170	16.460

TEMPERS AND LENGTHS

Type K } Hard Temper in straight 12 ft and 20 ft lengths.
Type L } Soft Temper in 30 ft, 45 ft, and 60 ft coils.
Type M only—Hard Temper in straight 12 ft and 20 ft lengths.

RECOMMENDED OPERATING PRESSURES

Type K—*Hard Temper* ..... up to 400 lb
Type K—*Soft Temper* ..... up to 250 lb
Type L—*Hard Temper* ..... up to 250 lb
Type M—*Hard Temper* ..... up to 250 lb

Revere Brass Pipe

Revere Brass Pipe is made in yellow brass for normal water conditions and red brass (85% copper) for unusually corrosive water conditions. It is preferable for water condenser lines for air conditioning, heating lines and process work in industry.

Herculoy

What It Is—HERCULOY is a silicon bronze, which when finished has a rich golden color. *It combines high strength and excellent physical properties with a high resistance to corrosion.* Its strength is comparable to that of the low and medium carbon steels, and its corrosion resisting qualities approximate those of pure copper. Herculoy is non-magnetic. It may be worked hot very easily. This exclusive Revere Product, on which U. S. Letters Patent have been issued, is an ideal engineering bronze.

HERCULOY posssses a simple, single-phase structure; similar to that of alpha brass, thereby permitting through proper annealing a range of physical properties intermediate between the limits set for tensile strength and elongation.

Applications—HERCULOY is recommended for Bolts, Nuts, Screw Machine Products, Turn Buckles, Lag Screws and accessories, in which forms it will be found to have greater strength than Steel with non-rusting and corrosion resisting qualities. For Piston Rods and Shafting where a superior article is required, Herculoy is supreme. Storage tanks for hot water made of Hot Rolled Herculoy Tank Plate have similar qualities to Sheet Copper Tanks plus an added strength equal to that of Steel. For electrical construction the great strength, non-magnetic property, toughness and resistance to abrasion possessed by Herculoy indicate its great value wherever suitable.

Physical Properties—The tensile strength of Herculoy in sheet form will vary with the amount of work and heat treatment given the metal during its production. The ultimate use to which the material is to be put determines the treatment given the metal in the course of its production.

Material	Tensile Strength (lbs. per sq. in.)	Elongation (in 2 inches)
Hot Rolled Sheets	68,000	51%
Cold Rolled Sheets	85,000	30%
Cold Rolled Sheets, Light Annealed	75,000	56%
Cold Rolled Sheets, Full Annealed	60,000	80%

Uses

Hot Water Storage Tanks
High Pressure Tanks
Kettles, Boilers and Mixing Bowls
Screen Plates for Process Industries
Pickling Tanks and Baskets
Blowers and Fans
Ventilating Ducts
Skylight Frames
Chemical and Sewage Plant Gates
Deep Drawing and Stamping

Revere Copper Sheet

Revere Copper sheet is the ideal material for air conditioning ducts where corrosive gases are drawn into the intake ducts from near by sources of pollution such as chimneys, and exhausts from photographic and battery establishments, garages, etc., or where ducts pass through rooms containing corrosive fumes and under some conditions Leadtex, a lead coated copper sheet, is advised.

Revere Concealed Radiation

Revere manufactures two distinct types of convectors: the Robras, which is the box fin type, and the Rocop, tubular convector, which meet practically all conditions of service, price, etc., where the use of the convector type of heater is considered. Full details will be gladly sent upon request. Address Revere Copper and Brass Incorporated, Rome Manufacturing Company Division, Rome, N. Y.

Revere Cooperation

To Heating and Air Conditioning Manufacturers—Revere Engineers are anxious to cooperate with manufacturers of heating and air conditioning equipment and help them solve their particular problems.

Revere Copper *and* Brass INCORPORATED

NRA CODE

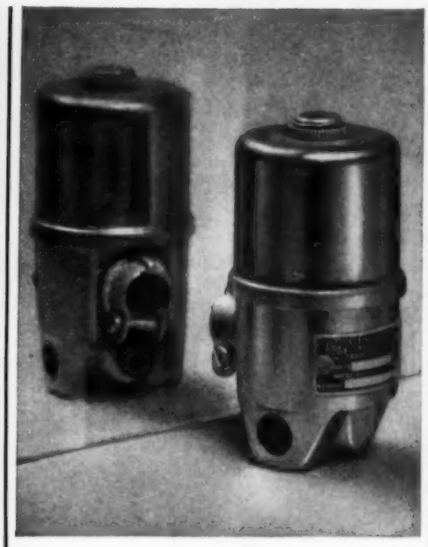
EXECUTIVE OFFICES: 230 PARK AVENUE, NEW YORK CITY · MILLS: BALTIMORE, MD. · TAUNTON, MASS.
NEW BEDFORD, MASS. · ROME, N. Y. · DETROIT, MICH. · CHICAGO, ILL. · SALES OFFICES IN PRINCIPAL CITIES

DEPENDABLE ELECTRIC ACCESSORIES BUILD YOUR PROFITS

The RIGHT MOTOR for FANS and BLOWERS QUIET . . . CARE-FREE . . . EASILY INSTALLED

For belt-driven fans and blowers requiring up to 1/4 hp., you can't go wrong if you select the G-E Type KH split-phase motor (illustrated).

It's dependable. It's easily installed because of its simplified, built-in terminal box, and because of its slotted base, which reduces the adjustment of belt tension almost to a matter of seconds. Add to these a lubrication system that provides *once-a-season* oiling, drip-proof end flanges that keep maintenance *down* by keeping undesirable materials *out*, and cushioned-power rubber mounting, and you have an equipment "buy" that's built to last.

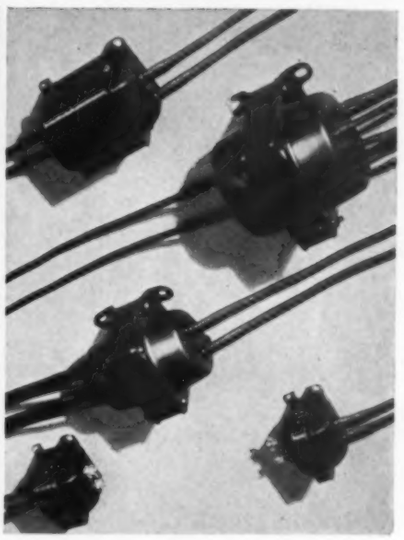


NEW, electrically operated VALVE SMALL . . . COMPACT . . . LOW-PRICED

Ideally suited to air-conditioning needs, this new electric valve will handle water or low-pressure steam quickly, easily, and economically, by remote control.

It provides positive operation over a wide range of pressures and temperatures. It is economical—it uses only 7 1/2 watts at 110 volts, 60 cycles. Its parts are sealed in steel —no stuffing box. A handy, built-in terminal box saves installation expense. A convenient metering pin provides 50-per-cent-flow regulation —no extra valve is needed.

If you have a large task for a little valve, this valve will save you money.



CASH IN ON THE COMPLETE LINE OF G-E CONTROL TRANSFORMERS

Small solenoids, relays, motors, lamps, and similar accessories needed for an air-conditioning installation often must be operated at various voltages—stepped down from the lighting circuit for safety, convenience, and simplicity of wiring.

There are standard G-E control transformers for many applications; your requirements for special voltages can be economically and quickly met from an extensive line of standard parts. May we send you information about these transformers?

To maintain and protect your reputation for service and dependability, it will always pay you to insist on the highest quality when buying electric equipment. For quality and correctness of design, order General Electric wire and wiring devices along with your motors, control, etc.

GENERAL ELECTRIC

General Electric Company
Dept. 6A-201, Schenectady, N. Y.

Please send me the following publications, as checked:

Name.....

Firm.....

Street.....City.....State.....

☐ GEA-1276A, Type KH Motors

☐ GEA-1847A, Solenoid Valves

☐ GEA-1358, Control and Signal Transformers

☐ Also, please place me on your mailing list to receive regularly your information service covering small motors, controls, transformers, etc., for air conditioning.

070-80

BYERS WROUGHT IRON

should be specified for Corrosive Service in all-welded systems

● Regardless of the method of pipe fabrication you employ, be sure to specify Genuine Wrought Iron. Especially, for the all-welded system, your specifications should read —"Byers Genuine Wrought Iron Pipe and Welding Elbows made from Byers Genuine Wrought Iron Pipe." Thus you are assured the long-life and economy of wrought iron in your all-welded system.

For further information on wrought iron, its use in all-welded systems and for service records which prove its long life and economy, ask a Byers Engineer or write our Engineering Service Department.

BYERS GENUINE WROUGHT IRON PRODUCTS

A. M. BYERS COMPANY
Pittsburgh, Pa. · Established 1864

Plant Heating—

[*Continued from page 25, front section*]

could make for higher maintenance expense than in the direct system, is the heater itself, with its accompanying motor and fan. This factor is of vital importance in choosing the particular type of heater for the plant. In making such choice, engineers and owners should consider carefully the merits of the units selected as regards design, construction and practicability for dependable performance.

In checking waste in already existing heating systems there are a number of considerations, says Mr. Moulder. A system can be intelligently checked only by calculation of the heat losses and consideration of the proper distribution of heating surfaces to overcome these heat losses. Frequently it will be found that existing plants have been changed to accommodate new manufacturing facilities and that while the old heating system may have been adequate for the original conditions, it is obsolete for the new. The advantages of unit heating as outlined above are equally true for existing systems, plus additional advantages.

It is furthermore advisable to check pipe sizes, pitch of piping, return systems and methods of venting. It is also well to check into the condition of insulation of various piping. Frequently a line is run through very much exposed area or portion and it would be beneficial if it were insulated. Insulation already installed may be in poor shape and in need of renewing. Underground piping should be thoroughly checked in that frequently the conduit or protection originally provided is deteriorated to a point where it is no protection at all and therefore very wasteful. Last but by no means least, all traps certainly should be checked thoroughly to make certain that they are in proper shape.

It also should be pointed out that each installation is a problem in itself and cannot be prescribed for by blanket statement.

Selection and Maintenance—

[*Continued from page 22, front section*]

will suffice. Local experience alone will dictate the time interval.

Instrument gaskets and bellows should be changed at least once every three years where an important process is involved and continuous operation is imperative.

Once a year the entire plant should be shut down, thoroughly dried out, carefully examined, all rust removed, all metal surfaces cleaned and painted and all insulation examined and repaired and the plant otherwise put in the best of condition.

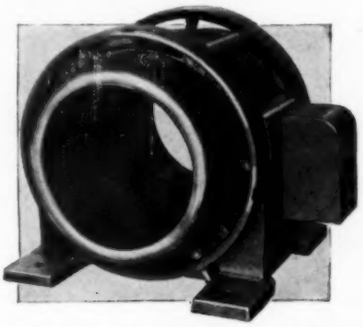
Air conditioning equipment so purchased, designed and maintained, will last twenty or twenty-five years, will cost little for repairs or replacements and will run continuously. Under such conditions it should make an attractive return and should prove a source of satisfaction to the purchaser.

Equipment Developments

For your convenience in obtaining copies of bulletins, see coupon on page 135

New Sealed Motors

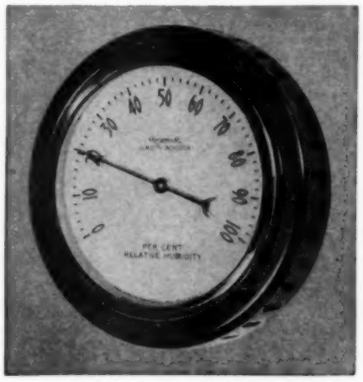
No. 136.—New "Seal-Clad" squirrel cage motors are of open type construction, with permanent coil protection. Hard, smooth, bakelite shields are sealed over the stator coils giving protection against metallic dust, grit, oil, moisture, mild acids and other agents injurious to insulation. They are build in ratings



up to 25 hp, 1800 rpm and suited to a much broader range of application than the ordinary open motor, states the maker. The motors have cast steel frames; twistless and distortionless stators; silver-brazed, indestructible rotors; oil and dust tight sleeve or anti-friction bearings.—Allis-Chalmers Mfg. Co., Milwaukee, Wis.

Humidity Indicator for Industrial Use

No. 137. New industrial humidity indicator designed for use in factories, warehouses, stores, etc., as an aid in manufacturing, finishing, drying, storing, curing packing, or processing gives direct reading in per cent relative humidity. Manufacturers state it is more accurate than wet and dry bulb method of obtaining relative humidity, requires no attention, is more convenient than a sling psychrometer.



Dimensions are 4⅝ in. by 1½ in. deep; stamped metal case and back; chromium bezel; convex glass crystal; cream colored metal dial with black figures; black indicating hand. Scale reads from 0 to 100 per cent. Instrument is of wall mounting type.—Fee and Stemwedel, 219 W. Chicago Ave., Chicago, Ill.

Flux for Brazing Metals and Alloys

No. 138.—New flux (called "Handy Flux") in paste form, has recently been announced. Developed to speed up and improve brazing operations on either ferrous or non-ferrous metals. It

TAYLOR FORGE WELDELLS

combine WROUGHT IRON'S long life with Correct Engineering

● With the increasing use of all-welded wrought iron systems, correctly designed fittings speed the work and improve the finished appearance of the installation. Therefore, it's important that you specify Taylor Forge Weldells made of Byers Wrought Iron Pipe.

This will assure you the long life of wrought iron plus the many advantages of Taylor Forge Weldells —i. e., correct engineering, selective reinforcing for greater strength, smooth interior and exterior, ends machine beveled for neat welding, tangents for accurate lining up. Write for suggested specifications, estimating prices and complete information.

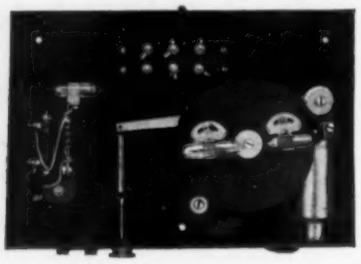
TAYLOR FITTINGS FOR WELDING FORGE

TAYLOR FORGE & PIPE WORKS
General Offices and Works, Chicago, Ill., P. O. Box 485
New York Office, 50 Church Street

has a lower melting point and permits taking full advantage of silver solders and brazing alloys having low flow points. Greater solvent action on a wide variety of oxides is said to be a feature that speeds up brazing and insures thorough wetting of joint surfaces at low brazing temperatures making it possible to obtain very strong and uniformly dependable joints. Recommended for brazing stainless steel, steel, monel metal, nickel, copper, brass, bronze and various other ferrous and non-ferrous metals and alloys.—Handy & Harman, 82 Fulton St., New York City.

A Water Level Control

No. 139. A water level control which may be used to start and stop a pump on systems with an elevated tank nearby or at a distance from the pump, operate high and low water alarm signal system or bells, operate low water signal and automatically stop pump when at high level (pump being manually started), or operate electric valves, is illustrated here (the cover is removed—cover is hinged at bottom and has circular window in front of disc).



In operation the straight line thrust of the metallic bellows type diaphragm is converted to a rotating motion of the disc carrying one or more mercury switches; the pressure applied to the diaphragm is counterbalanced by a weighted pendulum attached to the disc.

This control is a complete self contained single pump control requiring no auxiliaries, states the manufacturer.—Water Level Controls Company, 765 Hampden Ave., St. Paul, Minn.

Improved Pipe Union

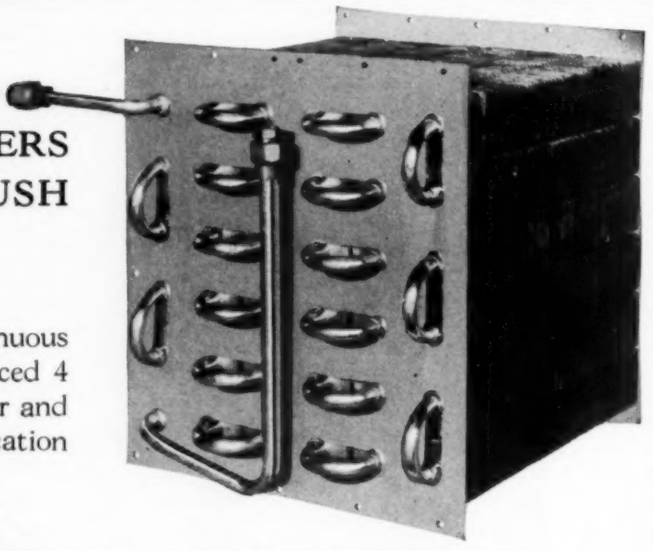
No. 140. The improved "Rockwood" union was developed to meet modern piping requirements, which have changed radically during the past few years. The union embodies a number of features, according to the maker, among which are an increase in strength as a result of the cold forging process used in the manufacturing; non-corrosive seat ring construction—silicon bronze for general requirements, and chrome iron for higher pressures and temperatures, and special corrosion resisting applications; seat rings double-locked to prevent working loose by expansion and contraction with temperature changes; porosity eliminated by cold working; parts interchangeable, so that if one part is damaged the entire union need not be discarded; and sherardizing to protect the union against corrosion.—Rockwood Sprinkler Company, 34-56 Harlow St., Worcester, Mass.

New Arc Welders and Electrodes

No. 141. A line of improved, portable, single operator arc welding sets incorporating several refinements has been developed. Self-stabilization is the outstanding feature, states the maker—an advantage which provides excellent performance throughout the entire welding range, using any type of electrodes, bare, lightly fluxed, or heavily coated.

Each welder is a self contained unit, having no external reactor, resistor, or separate exciter, and it is largely due to the self excited design of these machines that the inclusion of suffi-

FAN TYPE COOLING COILS BUILT TO CUSTOMERS SPECIFICATIONS BY BUSH



The Coil illustrated is made of continuous tubing, the fins are 2″ square and spaced 4 per inch. Both fin and tubes are copper and the entire unit tin dipped after fabrication

Certain applications might require a smaller fin and closer spacing in which case our 1⅝″ square fin spaced 6 per inch could be used.

There are now 38 combinations of fin and tube sizes and fin spacings in our standard line of finned tubing.

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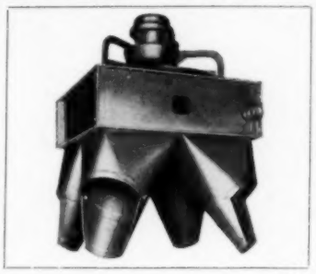
Wing Featherweight Heater



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FOR THE FOUNDRY

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It supplies uniform heat in every nook and corner and at every working level. The last word in scientific heating—(floodlight heating).

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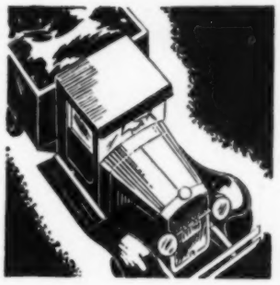

a heavy duty **TILE CONDUIT** *for Steam Lines run* UNDER TRAFFIC

WHERE conditions require great supporting strength under heavy traffic, Ric-wiL now offers its new special Super-Strength Tile Conduit. Note the extra heavy walls of these sections, also the special heavy-duty reinforcing at top, bottom and sides, giving balanced construction and maximum strength. Exhaustive laboratory tests indicate sufficient strength to support a concentrated *static 6-ton traffic load per wheel* under actual installation conditions. Thus any reasonable specifications on traffic load resistance can be met, regardless of conduit sizes. Write for Bulletin A-3412. New 1935 Bulletin of Heating Rates also sent on request.

The RIC-WIL Co., 1562 Union Trust Bldg., Cleveland
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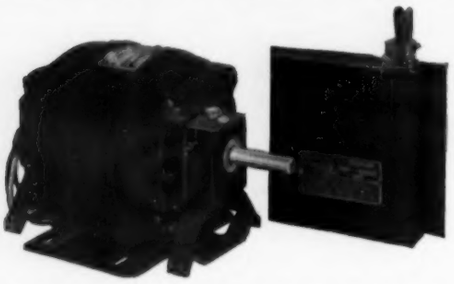
REGISTERED IN U. S. PATENT OFFICE
RIC-WIL
CONDUIT SYSTEMS FOR UNDERGROUND STEAM PIPES

For

CENTRIFUGAL TYPE FANS

This radial steel spring mounted motor for operating multiple fan air conditioning, heating, and ventilating units is available with extra long double shaft extensions for mounting a number of fans on each end.

Motor is of the low torque, single phase, capacitor type for adjustable speed operation—with built-in control accessories already wired up—or separately mounted if desired. A simple snap switch will select 3 speeds out of a possible 6.



For Propeller Type Fans

Spring mounted low torque, single, 2 and 3-speed capacitor motors—or single and 2-speed split phase motors—equipped with thrust bearings.

Adjustable Speed Motor with Separate Control for Belt Driven Fans

For belt driven fans higher starting torque motors are necessary. Overload circuit breaker protection is built in the control unit.

In addition to using a steel spring mounting to eliminate torque vibration, other inherent noises are removed by further refinements in electrical and mechanical design. Extra large bearings and oil reservoirs provide long life, with a minimum of attention.

Proper Regulation of air movement is an essential function of any air conditioning or ventilating equipment. Holtzer-Cabot pioneering is primarily responsible for the development of adjustable speed silent operating motors to successfully accomplish this function.

SINGLE PHASE — THREE PHASE — DIRECT CURRENT

MOTORS

Our engineers will gladly confer with you. Their experience can be helpful—write Dept. 20 for descriptive bulletin.

THE HOLTZER-CABOT ELECTRIC Co.

125 AMORY STREET, BOSTON, MASS.

Motor Specialists for 50 Years

cient self-stabilization to provide steady welding characteristics under all conditions is possible.

Separate controls are provided for adjustment of welding current and welding voltage. These controls are located on a dead-front panel on top of the machine at convenient height above the floor. On this panel are also located a switch handle for reversing polarity and an instrument with selector switch for indicating welding current and welding voltage. All controls are clearly labelled to facilitate rapid and accurate adjustment.

All working parts of the machine are protected against damage from water or other foreign materials by completely drip-proof construction. The entire set is low-hung on a three-wheeled truck which provides portability and mechanical stability; the set may safely be tipped as much as 22 degrees from the vertical.

This manufacturer has also announced a new heavily-coated arc-welding electrode for the economical production of high-speed, high-quality welds in the flat position. These "Type W-23" electrodes are suitable for either manual or automatic arc welding and will produce equally good results on either alternating or direct current. With the latter, reverse-polarity is used.

These electrodes are recommended for pipe welding and other purposes.—General Electric Co., Schenectady, N. Y.

Smoke Alarm Promotes Economy

No. 142. A recently developed smoke alarm is a simplified indicating device for signaling the boiler fireman when smoke conditions in the stack become excessive. When smoke enters the stack, this alarm unit warns the fireman to check his combustion instruments instantly and adjust his drafts to insure maximum operating efficiency.

It not only permits adherence to smoke abatement laws, but also eliminates fuel waste reflected by excessive smoke conditions. In the Rex Cole Building, New York City, a smoke alarm unit was installed in 1933 and during the unusually severe winter of 1933-34, the monthly fuel costs averaged 22 per cent less than during the previous winter. The original investment in this unit has been repaid several times during its first year of operation in this building according to the manufacturer.

The alarm is self-operating. A beam of light is focused through the smoke stack on to a sensitive "Photronic" photo-electric cell. This cell generates electricity in proportion to the amount of

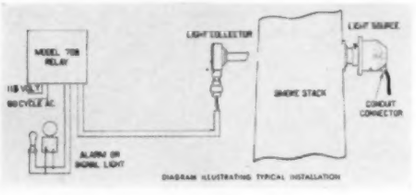


DIAGRAM ILLUSTRATING TYPICAL INSTALLATION

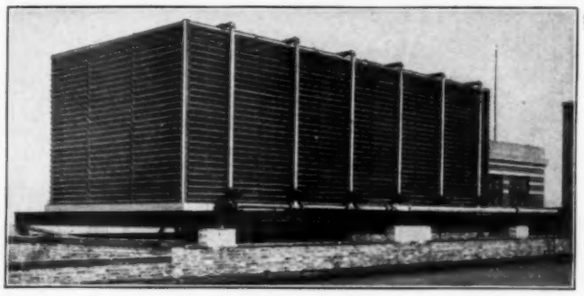
light which reaches it. Smoke reduces the intensity of the light on the cell and the corresponding loss in current generated permits a relay in the boiler room to operate an alarm or signal light.

The complete alarm consists of a light collector, a light source, and a relay panel. The light collector, which contains a photo-electric cell, is mounted on the side of the stack. The visor on the front of the collector is hinged so that the glass covering the cell may be readily cleaned. The cell housing has a ½" conduit fitting so that the leads may be run to the relay panel through a conduit which will serve as a support for the unit.—Weston Electrical Instrument Corp., Newark, N. J.

First Choice of Engineers

BINKS

Spray Cooling Towers



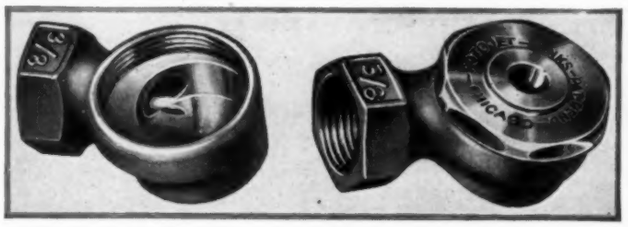
BINKS Atmospheric Spray Cooling Towers—built in sizes from 6 to 200 tons refrigeration capacity—as well as Indoor Forced Draft Cooling Towers up to 40 tons capacity, are now used extensively in connection with the rapid march of progress being made in the Air Conditioning field.

BINKS offers a standardized complete shop constructed Tower (with full spray nozzle distribution) for almost every application, as will be evidenced by more than 3,000 Binks Cooling Towers now in operation.

BINKS patented "ROTOJET" Clog-Proof Spray Nozzles likewise are contributing mightily to the increased Air Washing efficiency now in demand.

The Binks line—recognized as being the most complete—includes spray and atomizing nozzles of advanced design for practically every industrial application.

Valuable data on request.



ROTOJET Nozzles furnished in sizes from 1/4" to 2½" pipe connections. Nozzle with cap removed shows involute whirl chamber.

A Few of the 3000 Binks Users

Goodyear Tire & Rubber Co.
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Roosevelt Hotel Co.
University of Michigan
Red Top Malt Extract Co.
Buda Company
Georgia Power Co.
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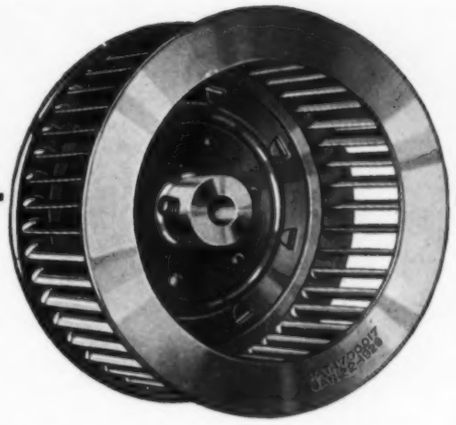

ALUMINUM BLOWER WHEELS

Attractive appearance.

Light weight minimizes power consumption.

No protective finish necessary.

Lower transportation costs because of light weight.

Aluminum construction minimizes resonance and wheel operates more quietly.

No rivet head or other projections on inlet face of wheel.

Patented construction reduces manufacturing cost.

Rugged design—tested to speed far in excess of normal.

Made from a special aluminum alloy having great strength.

Approximately one-third weight of steel wheels.

Low starting torque.

Send for Bulletin showing dimensions for full range of sizes 3" to 15" diameter, also drawing with dimensions for housing scrolls. Capacity rating sheets now available.

THE TORRINGTON MFG. CO.

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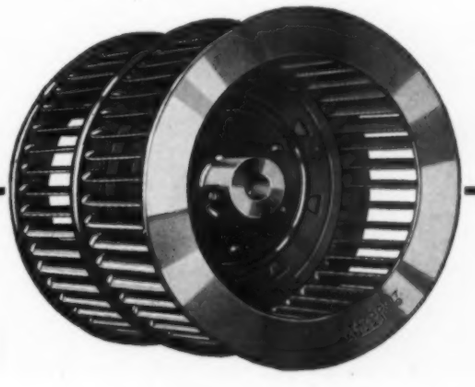

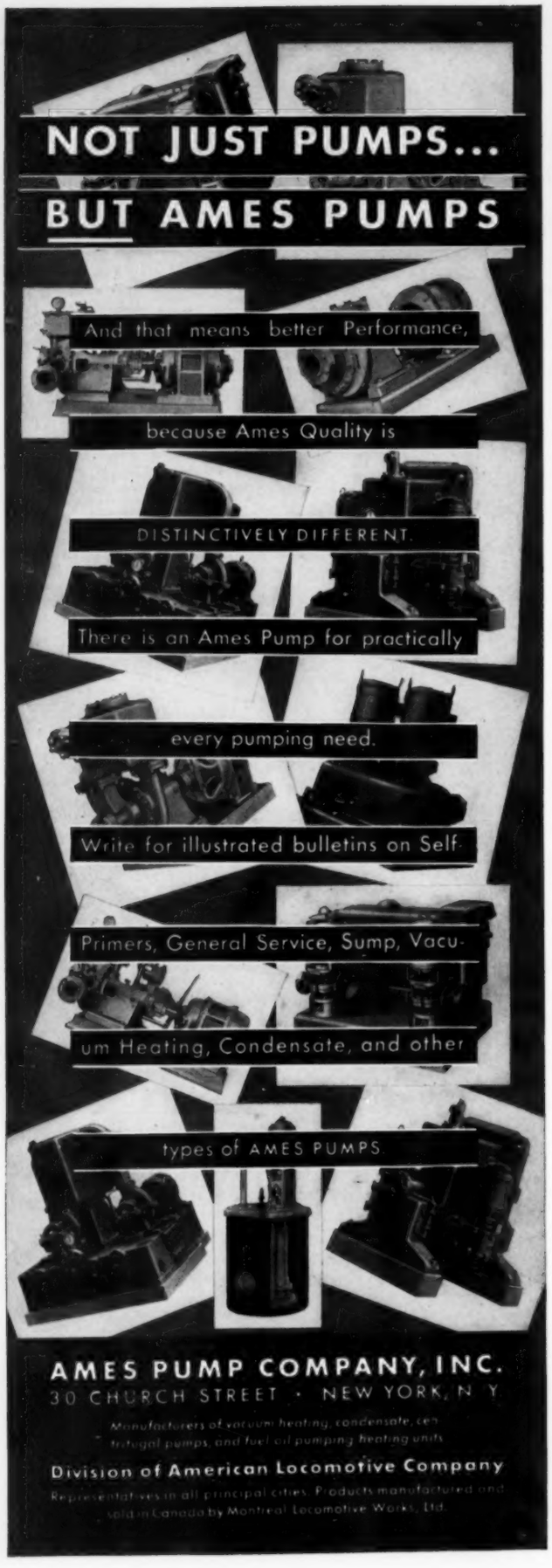
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sold in Canada by Montreal Locomotive Works, Ltd.

Stethoscope Detects Gas Leaks

No. 143. Principle involved in pocket size stethoscope for locating gas leaks is that a fluid escaping from a container sets up vibrations in the fluid within the container. The fluid then transmits the vibrations to the stethoscope which is attached to an opening in the container. The stethoscope chamber is so

constructed as to collect and amplify these vibrations. The amplified vibrations are then transmitted to the ear through the rubber tubes and binaulars.

In order to locate a leak of fluid from a pipe line system the stethoscope is attached to an opening. If there is a leak it will be heard as a hissing sound. The nearer the leak the louder it sounds. The smaller the pipe the further the sound will travel. —E. R. Gilmore, 214 E. First St., Topeka, Kansas.

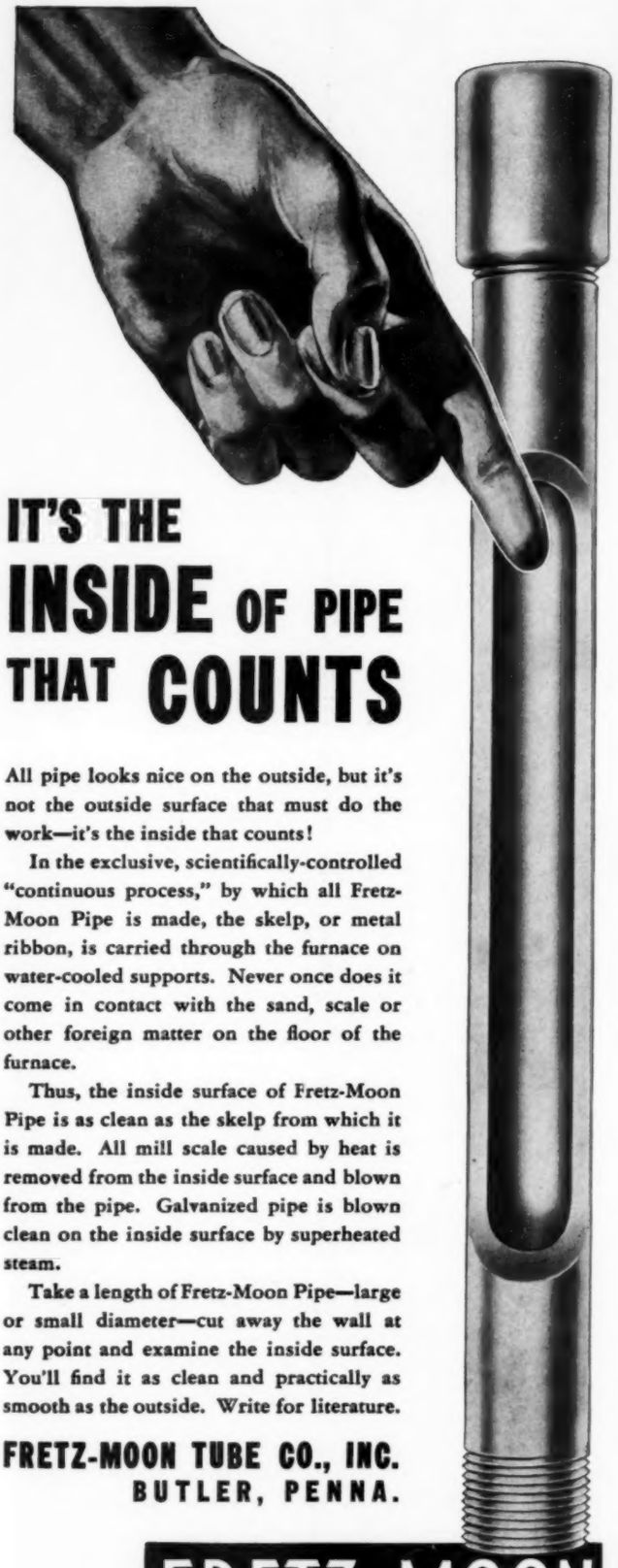
Goggles Have Double Use

No. 144. New eye protecting goggles combine protective features of chippers' and welders' types, eliminating need for changing goggles. The upper view shows the goggles as used in weld-

ing for protection against dangerous light rays, glare, etc., and the lower view for protection against impact, flying chips, etc.—Willson Products, Inc., Reading, Pa.

Clip Type Gate Valve

No. 145. A clip type gate valve with several new features has a round bonnet and differs essentially from the conventional style of clip gate valves. Round bonnet construction permits the use



IT'S THE INSIDE OF PIPE THAT COUNTS

All pipe looks nice on the outside, but it's not the outside surface that must do the work—it's the inside that counts!

In the exclusive, scientifically-controlled "continuous process," by which all Fretz-Moon Pipe is made, the skelp, or metal ribbon, is carried through the furnace on water-cooled supports. Never once does it come in contact with the sand, scale or other foreign matter on the floor of the furnace.

Thus, the inside surface of Fretz-Moon Pipe is as clean as the skelp from which it is made. All mill scale caused by heat is removed from the inside surface and blown from the pipe. Galvanized pipe is blown clean on the inside surface by superheated steam.

Take a length of Fretz-Moon Pipe—large or small diameter—cut away the wall at any point and examine the inside surface. You'll find it as clean and practically as smooth as the outside. Write for literature.

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MUNDET NATURAL CORK ISOLATION MATS, for machine vibration, will solve another problem that confronts the Air Conditioning field.

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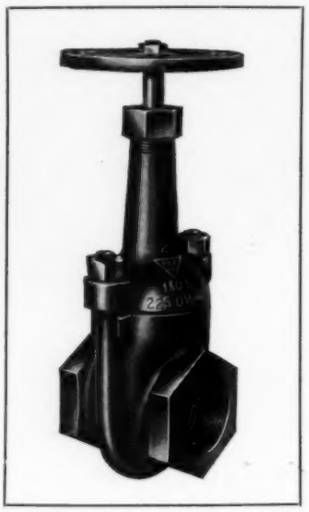
of a male and female joint with asbestos gasket assuring tightness, states the maker.

A full U-bolt is used to hold the bonnet joint tight. When the nuts on these bolts are tightened, pressure is equalized on both sides. Three set of guides on the body and two solid eye lugs on the bonnet hold the U-bolt and eliminate possibilty of slipping.

Among internal features are the full thickness I-beam cast



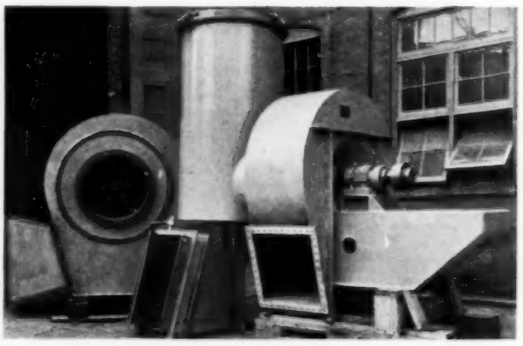
wedge which brings sturdiness to the valve seat and permits the use of full size body guides to assure correct seating. A horseshoe connection is used to assemble the spindle and wedge, providing great strength and lifting surface. Guide slots in the wedge are machined.

The valves are made in all-iron and bronze mounted construction.—Reading-Pratt & Cady Co., Inc., Bridgeport, Conn.

Acid-Proof Rubber Covering Applied to Fans

No. 146. The practicability of utilizing acid-proof rubber covering to eliminate destructive corrosion is exemplified in the accompanying illustration showing large exhaust fan units recently rubber covered.

This covering is a specially-developed acid-proof rubber com-



pound applied by a perfected process to the metal surfaces of the fan unit.

The process of uniting rubber to metal holds the rubber covering in place so tenaciously that after vulcanization, the cover cannot be removed without tearing and destroying it, says the maker. —The Manhattan Rubber Mfg. Division, Passaic, N. J.

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Comprehensive information for architects and engineers in handling cooling problems is now available in our new bulletin describing Aerofin Standardized Light Weight Fan System Cooling Surface.

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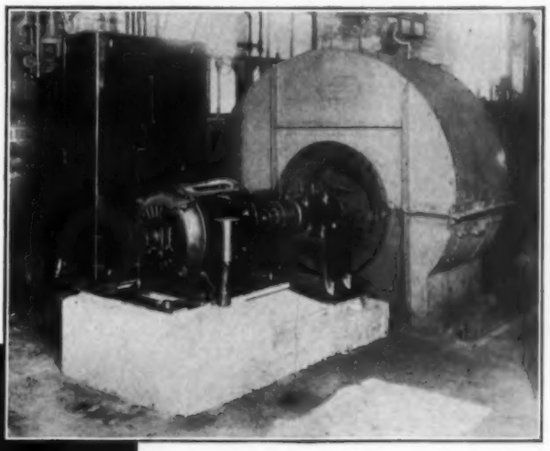
Multi-Speed *Motors for fans*



2 SPEED

3 SPEED

4 SPEED

CONSTANT TORQUE

CONSTANT H P

VARIABLE TORQUE

PICTURED above is a Wagner multi-speed motor driving a forced-draft fan in the boiler house of a large drug company. It is a 25/11/6.2 horsepower, 1725/1140/860 rpm variable-torque type MRP-1 motor.

When it's a multi-speed motor application — consult Wagner. 2-speed, 3-speed, or 4-speed—constant - torque, constant - horsepower, or variable-torque — open frame, drip-proof, or totally-enclosed fan-cooled — horizontal or vertical — sleeve or rolling bearing — ½ or 125 hp—whatever the motor requirement may be, there's a Wagner multi - speed motor designed and built for the job.

The motor illustrated above as well as other types of Wagner multi-speed motors, is described in Bulletin 174. Ask for a copy.

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Name and Position

Firm

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M333-1

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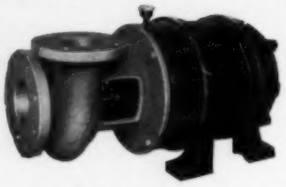

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Write for Bulletins or any other information desired. AMERICAN STEAM PUMP COMPANY, Battle Creek, Michigan. . . . . .

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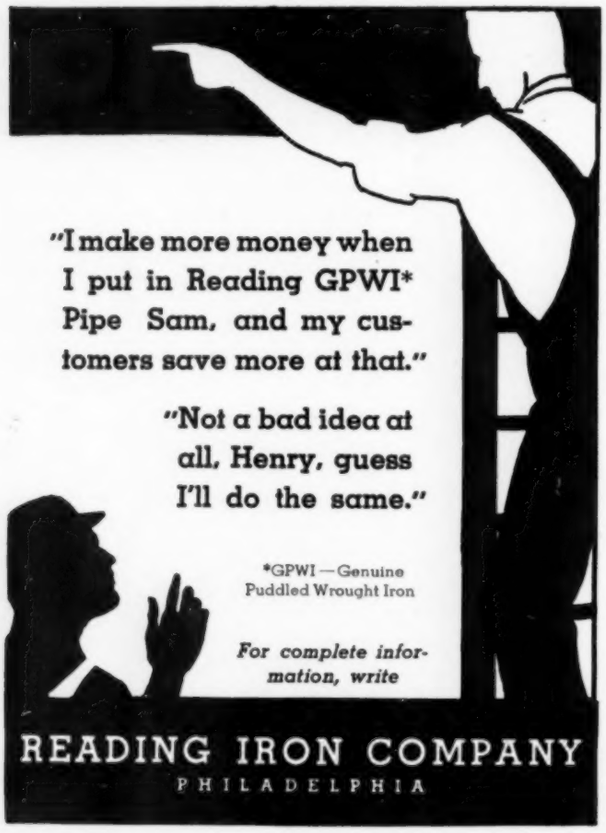
Centrifugal and Steam Pumps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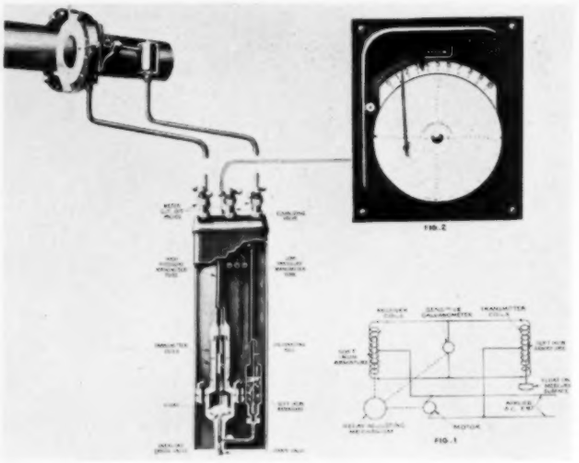
SCIENCE AND INVENTION HAVE NEVER FOUND A SATISFACTORY SUBSTITUTE FOR GENUINE PUDDLED WROUGHT IRON

Electric Flow Meter

No. 147. The electric flow meter illustrated here is distinguished by the fact that the power for the operation of pointer, pen and integrator is the same throughout the entire range of measurement. The pressure differential caused by flow through the orifice or nozzle is not required to move the registering mechanism, which is done instead by external power from an electric motor, the application of this power being controlled by a sensitive relay.

Flow through the pipe line causes a float carrying a soft iron armature and resting on the mercury in a U-tube to be displaced



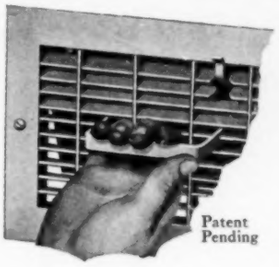
a certain distance and this displacement is balanced electrically by shifting a similar armature in the recorder, the two armatures being surrounded by the inductance coils of a bridge circuit traversed by an alternating electric current. Unbalance of the circuit deflects the boom of a sensitive galvanometer, which directs the application of power to shift the recorder armature until the galvanometer boom returns to neutral. The value recorded is that at which balance is attained, regardless of the direction from which the value has been approached and there is no lag caused by friction or other non-reversible forces. The circuits and galvanometer are so proportioned that a displacement of the transmitter armature of less than 0.001 in. causes the galvanometer boom to deflect and is, therefore, registered by the recorder, corresponding in the usual installation to but 0.01 of one per cent change in rate of flow at mid-scale.—Cochrane Corp., Philadelphia, Pa.

Multiple-Point Recording

No. 148. Reading of multiple-point instrument records is simplified by a new system of "numerals in colors." This marking system can be specified on "Micromax" strip-chart recorders for 2, 3, 4 or 6 points. With each of the thermocouples, resistance thermometer bulbs or other primary elements identified on the chart by a numeral, and each numeral distinguished from the others by being printed in a contrasting color, reading of records is made easy, errors are avoided and time is saved. Colors used for printing the numerals are: black, green, red, violet, yellow and blue.

The print-wheel on these multiple-point recorders has a separate inking pad for each point to be recorded.—Leeds & Northrup Co., Philadelphia, Pa.

Having Trouble Directing Air Circulation?



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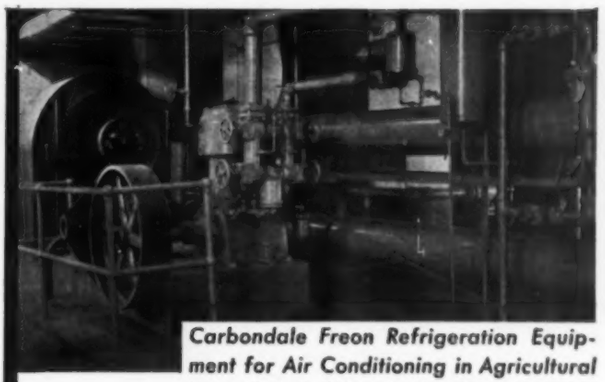

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Just one example of a complete engineering, equipment, and installation service, for any air conditioning requirement . . . and for EVERY TYPE OF REFRIGERATION

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PERSONAL

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When it comes to controls for air conditioning equipment, where can you secure better help than that available from air conditioning control specialists? For ten years, ALCO Valves have been the outstanding favorites with leading refrigeration engineers—and used in the majority of refrigerating installations for the control of gaseous and liquid refrigerants, water and brine.

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Indicating Relative Humidity Controller

No. 149. Accuracy of a membrane type humidity recorder is combined in a new instrument with "Rotax" type contacts to form an electrically-operated humidity controller that will give accurate control throughout its entire life, according to the maker.

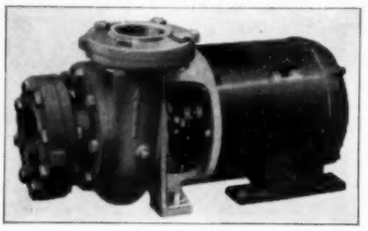
The controller can be located conveniently wherever electric current is available. It maintains humidity by operating solenoid valves and motor valves on water, air or steam lines. The relay set required to accomplish this is located inside the case. It can be used to operate unit heaters, control dampers, air washers and other air conditioning apparatus. The shape of the case is such that minimum space is required for mounting.

Standard contacts are used in this instrument. The range is 0-100 per cent relative humidity and it is designed for use between +10 F to +120 F.

Room or duct temperature controllers can be furnished also in the same style case for any industrial application. Interlocking electrical circuits can be furnished when required.—The Foxboro Co., Foxboro, Mass.

New Centrifugal Pump

No. 150. A compact, close-coupled centrifugal pump of high efficiency with capacities ranging from 100 to 600 gpm against heads up to 189 ft is ideal for circulating and cooling systems, standpipe and water supply systems in factories, warehouses, apartment buildings, homes, hotels, hospitals, office buildings and



for transferring all reasonably clear industrial liquids, states the maker. It is a single-stage, end-suction pump. The pump and motor are built together as one unit with a single shaft. The enclosed type impeller is keyed to the motor shaft, which extends into the pump casing. There is only the one moving part.—Chicago Pump Co., Chicago, Ill.

New Type of Suction Hose

No. 151. A new type of suction hose which has a woven tubular re-inforcement of strong cabled cotton cords and wire, making a strong but light hose which will not collapse, has recently been developed.

This new hose has a smooth long life lining of specially compounded rubber and offering mimimum frictional resistance to flow. Another advantage of this hose is that if it should accidentally be crushed, it can be quickly restored to shape by use of a vise or mallet.—Electric Hose and Rubber Company, Wilmington, Delaware.

Recent Trade Literature

For your convenience in obtaining copies of bulletins, see coupon on this page

No. 241. Expansion Loops: Tube-Turns, Inc., 405 Herald-Post Bldg., Louisville, Ky. 8 page booklet—with large chart—of engineering and design data for simplified design of expansion loops using "Tube-Turns" and straight lengths of pipe or tubing. Chart is based on modern bending theory. Suitable loops designed by varying lengths of pipe at top of loop or of the side legs, depending upon construction limitations.

No. 242. Gages: The Hays Corp., P. O. Box 299, Michigan City, Ind. 2 page bulletin describing line of electric contact gages for indicating and controlling draft, pressure, and differential draft or pressure. Uses include controlling static pressure in air distribution ducts; operation of dampers to maintain proper air flow through branches of a zone system; indication and control of abnormal resistance through air filters, dust collection systems, refrigerating coils, etc.

No. 243. Ignition Transformers: General Electric Co., Schenectady, N. Y. Data sheet for oil burner ignition transformers, giving detailed description, ratings, etc.

No. 244. Insulation: Johns-Manville, 22 E. 40th St., New York City. 12 page booklet on industrial roofs, including a discussion of roof insulation and its effect in reducing fuel costs, preventing condensation, and improving working conditions.

No. 245. Mechanical Rubber Goods: New York Belting and Packing Company, 1 Market St., Passaic, N. J. 96 page catalog of belting, hose, packings, and miscellaneous mechanical rubber goods including complete information on the products, helpful tables, drawings, formulas, and general information.

No. 246. Motors: Allis-Chalmers Mfg. Co., Milwaukee, Wis. 4 page leaflet describing new "Seal-Clad" induction motor with protected windings built in ratings up to 25 hp, 1800 rpm, ball or sleeve bearings.

No. 247. Noise and Vibration Control: Cork Insulation Co., Inc., 155 E. 44th St., New York City. Data sheet on isolation corkboard, a material developed for the control and elimination of noise and vibration and having a greater density than ordinary corkboard to withstand machinery loads. Uses and installation are discussed briefly.

No. 248. Pipe Thawing: Hobart Brothers, Troy, N. Y. Looseleaf mimeographed bulletin on thawing pipes by means of the Hobart arc welder containing complete discussion of the method and numerous "case studies" with diagrams, costs, etc. (Available without charge to those interested in this company's new 40 volt arc welders, 40c to others).

No. 249. Pumps: Worthington Pump & Mchry. Corp., Harrison, N. J. 4 page bulletin on centrifugal pumps, single stage volute type.

No. 250. Valves: The Fairbanks Co., 396 LaFayette St., New York City. 6 page folder describing renewable gate valves, with sizes, dimensions, and other data.

No. 251. Water Vapor Refrigeration: Ingersoll-Rand, 11 Broadway, New York City. 24 page booklet discussing very completely centrifugal water vapor refrigeration, including a description of the development of this type of machine, the problems involved, experimental units and tests, commercial units, applications, characteristics, etc.

No. 252. Welding: General Electric Co., Schenectady, N. Y. 12 page bulletin devoted to detailed description of G-E self-stabilized, single-operator arc welders. Also, 16 page booklet on welding electrodes and accessories, discussing in informative manner the choice of the proper electrode for successful welding, each type being considered in separate section.

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Please ask the manufacturer to send me more information about the equipment mentioned under the following reference numbers in "Interesting Equipment Developments" and "Recent Trade Literature." (Check numbers in which you are interested):

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143	144	145	146	147	148	149
150	151	241	242	243	244	245
246	247	248	249	250	251	252

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